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REFRIGERATION CLASSIFIED . 1

THE REFRIGERATING DATA BOOK

THE
REFRIGERATING DATA BOOK

BASIC VOLUME
Seventh Edition

THE AMERICAN SOCIETY OF
REFRIGERATING ENGINEERS

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PREFACE

This 7th edition (1951) of the REFRIGERATING DATA BOOK—Basic Volume continues the ASRE policy of endeavoring to present in its two DATA BOOKS the most important information fundamental to the theory, design, and broad application of refrigeration. The BASIC DATA BOOK includes primarily information on refrigeration design, theory, thermodynamics, refrigerant properties, and related topics, while in the Applications Volume emphasis is given to detailed discussions of the practice and art of refrigeration. The respective DATA BOOKS are published on a biennial basis. The current Applications Volume of the DATA BOOK is the 1950 edition and a revision of it is not scheduled until 1952. The Basic Data Book Volume preceding this edition appeared in 1949.

The present edition was not so extensively revised as was the previous edition which involved a complete resetting. This Volume includes a number of improvements in presentation, correction of minor errors, the rewriting of five chapters and partial rewriting of several more. Additional tabular information has been supplied to meet needs which were found to exist while the last edition was in use and to supply new data on refrigerants, insulating materials and practices which have come to the attention of the refrigeration industry during the last two years.

An effort has been made to reduce the over-lapping and duplication of subject matter approached from the varying backgrounds of different authors. It is hoped that this has been reduced to a minimum. This edition has again been prepared under the guidance of six associate editors. Although many of the chapter authors are the same as those in the previous edition, each one has had an opportunity carefully to review and modify previously written material. It can be seen that many new names appear in the list of more than fifty authors.

The fine work of Mr. C. E. Oberst, Technical Editor, in checking, arranging and carrying out detailed compilation on this Volume is much appreciated by other members of the editorial staff. Mr. Oberst continues in the fine traditions which Miss Julia Szabo developed in the production of previous DATA BOOKS.

Suggestions by readers as to items requiring clarification or inconsistencies will be welcomed by the Society. This work represents the unselfish efforts of a large number of cooperating individuals and all of the authors merit the thanks of the Society and of the refrigerating industry for the generous contribution of time and ability which they put into making this book possible.

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PART I
THEORY

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1. BASIC THERMODYNAMICS

Definitions of Terms

1. **Refrigeration** is the science of producing and maintaining temperatures below that of the surrounding atmosphere. Low temperatures may be produced in a number of ways, the one most frequently used being that of allowing an expansion of a fluid to take place in such a manner as to extract heat. Consideration of the simplified schematic diagram of a compression refrigerating system shown in **Fig. 1** will lead to an understanding of what thermodynamic changes are involved. The system may be considered as having a low-pressure and a high-pressure side; this separation is shown by the broken line. Low-pressure vapor (refrigerant) from line (1) enters the compressor (A) and is compressed to a high pressure. Passing through line (2), the high-pressure vapor enters the condenser (B), where it is condensed under constant pressure by some medium (usually air or water). The high-pressure liquid in line (3) then expands through the throttling valve (C) to a lower pressure in the evaporator (D). The latent heat necessary to evaporate the refrigerant is taken from brine being circulated through coils in the evaporator, thus cooling the brine. The low-pressure vapor is then ready to be compressed and repeat the cycle, which is indicated on the skeleton Ph chart, **Fig. 2**.

The change 1–2 is an isentropic (*s* const) compression and 2–3 a constant pressure contraction, the superheat being removed from the refrigerant (2–2′) and condensation taking place (2′–3) while the latent heat of vaporization is being removed. The change 3–4 represents throttling (*h* const) of the refrigerant through the expansion valve and 4–1 absorption of heat by the refrigerant in the evaporator.

The absorption-type refrigerating system relies upon the expansion of a high-pressure liquid and its subsequent evaporation at low pressure to produce cold in exactly the same manner as the mechanical compression system. These systems differ, however, in the manner in which the liquid is put under high pressure.

2. **Thermodynamics** in a broad sense is the science which treats of energy. In a more restricted sense thermodynamics is the science which treats of heat energy and its transformations into other forms of energy. Thermodynamics develops the laws governing these transformations, and studies the properties of the various media by which the transformations are effected. Finally, the principles thus developed are applied to the various heat machines.

3. A body or system of bodies is said to possess **energy** when capable of doing mechanical work while undergoing a change of state.

Kinetic energy is the energy a body possesses by virtue of its motion, the amount of which is determined by

$$\tfrac{1}{2}\, m \text{ vel}^2$$

where *m* is the mass and vel the velocity of the body. For a system of bodies, the total kinetic energy is found by adding the separate kinetic energies of all the bodies making up the system.

Potential energy of a system is due to its position or configuration. The work the

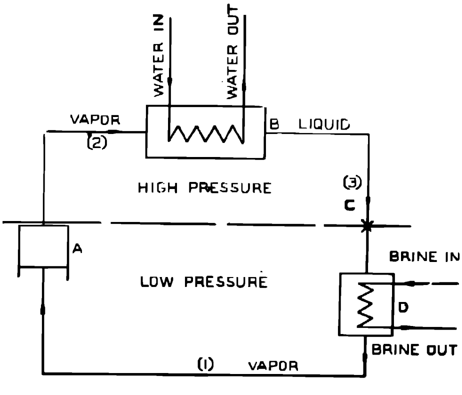

Fig. 1. Scheme of Compression Cycle

system can do in passing from its given position or configuration to a standard reference position or configuration is the measure of its potential energy. The hammer of a pile-driver, when raised, possesses potential energy due to its position, while a stretched rubber band possesses potential energy due to its configuration.

4. **Work** is accomplished when a force acting on a body displaces it. The amount of work done by the force is determined by the product of the magnitude of the force and the distance through which the body is moved. Thus, if a force of 10 lb displaces a body 5 ft, the amount of work done is $10 \times 5 = 50$ ft lb. The work done by an expanding fluid is given by, work $= \int_{v_1}^{v_2} P dv$. This integral can be evaluated only if the pressure, P, is known as a function of the volume, v. v_1 and v_2 are the initial and final volumes, respectively, of the fluid.

5. **Heat,** from a strictly thermodynamic standpoint, is **thermal energy** in transition. When heat flows into a body, it is stored as thermal energy in the body. The unit used to measure heat and thermal energy is the British thermal unit. The mean Btu is defined as 1/180 of the amount of heat necessary to raise the temperature of one pound of water from 32 F to 212 F under a pressure of one standard atmosphere.

The **specific heat** of a substance is the amount of heat, expressed in Btu, required to raise the temperature of one pound of the substance one degree Fahrenheit.

The **normal latent heat of vaporization** of a substance is the amount of heat, expressed in Btu, required to change one pound of the liquid to vapor at standard atmospheric pressure. The temperature, being the saturation temperature corresponding to the particular pressure of the substance during vaporization, remains constant during the change. The latent heat of vaporization will vary with pressure. Since the change takes place at constant pressure, the latent heat of vaporization is equal to the change in enthalpy.

The **latent heat of sublimation** of a substance is the heat necessary to change one pound of the solid substance to vapor without passing through the liquid state. The latent heat of sublimation of CO_2 is 248 Btu per lb. This, together with the approximately 27 Btu which is absorbed by the vapor in going from −109 F, the subliming temperature of CO_2 at atmospheric pressure, to 32 F, makes a total heat-absorbing capacity of 275 Btu per lb.

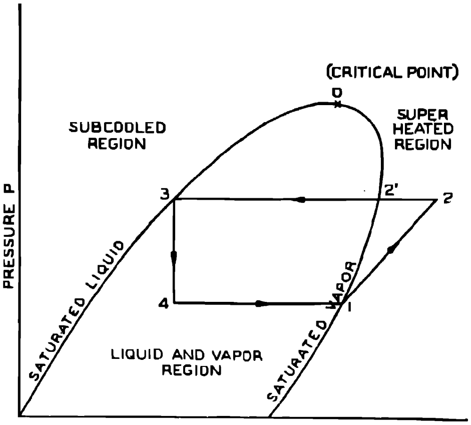

Fig. 2. Pressure-Enthalpy Diagram

The **latent heat of fusion** of a substance is the amount of heat necessary to change one pound of the solid to liquid, the temperature and pressure remaining constant during the change.

Thermal energy may be transferred as heat by any one of three methods or combinations thereof, namely, **radiation, conduction,** or **convection.**

A body may dissipate internal kinetic energy by emitting electromagnetic radiation in the form of waves. This process of transferring internal energy from one body to another is called **radiation.** These electromagnetic waves are identical in character with light and radio waves, differing from them only in wave length.

Conduction in solids is probably effected by a transfer of kinetic energy between adjacent vibrating molecules.

Convection is the transfer of thermal energy by mass movements. The movement of warm air through ducts, whether forced by a fan or free due to density variations, is by convection. Cooling a condenser by air is partially a convection process. Heat flows by conduction from the condenser walls through a thin stagnant film of air, and is finally convected away by the moving air which is in contact with the stagnant film.

6. The **working substance (refrigerant)** is the medium by which the necessary transformations are carried out during the refrigerating cycle. From a thermodynamic standpoint, a number of characteristics are desirable in the working substance. The saturation pressure corresponding to the condenser temperature should be low, while the saturation pressure corresponding to the desired evaporator temperature should be high, since the smaller the pressure ratio the less the compression work. The vapor should have a high specific heat, while that of the liquid should be low. The specific volume-specific enthalpy relationship should be such that the required piston displacement for a given load will be small. The latent heat should be large. The coefficient of performance should be as large as possible and the saturated vapor line should approach as nearly as possible an isentropic line.

Refrigerant Properties

7. **Properties** of the refrigerant which are important in engineering thermodynamics are six in number. Three of these, **pressure, volume** and **temperature,** are directly measurable. The other three, **internal energy, enthalpy** and **entropy,** are functions of the first three.

Pressure, P, is expressed in lb per sq ft or psi.

$$P_{abs} = P_{gage} + 14.7 \text{ psi}$$

Specific volume, v (per lb), the volume occupied by one pound of the substance, is expressed in cu ft per lb.

Temperature, T, is expressed in degrees Fahrenheit absolute.

$$T_{abs} = t_{Fah} + 460$$

From the Carnot theory, the temperature of a body is said to be zero absolute when all of the thermal energy has been removed from the body. This definition of absolute zero does not presuppose any theory of heat. From the standpoint of kinetic theory, absolute zero is reached when all molecular motion ceases, which, of course, presupposes a theory of heat.

Internal energy, u, is expressed in Btu per lb. Kinetic theory attributes the internal energy of a substance to the motion and configuration of the particles (molecules, atoms, and sub-atomic particles) which go to make up that substance. That part of the internal energy due to the motion of the particles is termed sensible thermal energy, while that part due to the configuration of the particles is termed latent energy. Only the sensible thermal energy can be detected by a thermometer.

Enthalpy, h, is defined by the expression $h = u + APv$, where $A = 1/778$. In a flow system Pv is called flow work. When steady continuous flow exists, the product Pv represents the energy necessary to inject the fluid into the pipe entrance. Although in the case of a non-flow system no physical meaning can be attached to this term, the expression for h is still useful. For a non-flow system the change in h is equal to Q under a constant pressure change.

Entropy, s, is expressed in Btu per deg F·lb. For an orderly process (reversible), the change in specific entropy is defined by the expression $ds = dq/T$. The use of this term will be made clear in the discussion of the Carnot cycle (p. 9).

8. The **graphical representation** of properties is frequently used in the solution of refrigerating problems. Any two properties may be used as rectangular coordinates for a chart.

A skeleton Ph chart of the type most frequently used in refrigerating work is shown in Fig. 2. The saturated liquid line is the locus of the state points which represent liquid having a temperature corresponding to the saturation pressure. The saturated vapor line is the locus of the state points which represent vapor having a temperature corresponding to the saturation pressure. The two lines meet at 0, the critical point. The critical temperature is that temperature below which a gas must be cooled before it can be liquefied, regardless of the amount of pressure applied. In the area to the left of the saturated liquid line, called the subcooled region, the temperature of the liquid is less than the saturation temperature corresponding to the pressure of the liquid. In the area between the saturated liquid and saturated vapor lines a mixture of vapor and liquid exists. In the area to the right of the saturated vapor line, known as the superheated region, the temperature of the vapor is greater than the saturation temperature corresponding to the pressure of the vapor. A state point is determined by any two properties on the chart except in the wet region, where the quality must be given in addition to the two properties.

The *Pv*, *Ts* and *hs* charts are also used in refrigerating work. The *Pv* and *Ts* charts have the advantage that areas on these charts have physical meanings. Since the work done by the expanding fluid is given by work $=\int_{v_1}^{v_2} P\,dv$, the area under the curve on the *Pv* diagram represents work. From the definition of change in entropy, $dS = dQ/T$, it is seen that areas on the *Ts* diagram represent heat.

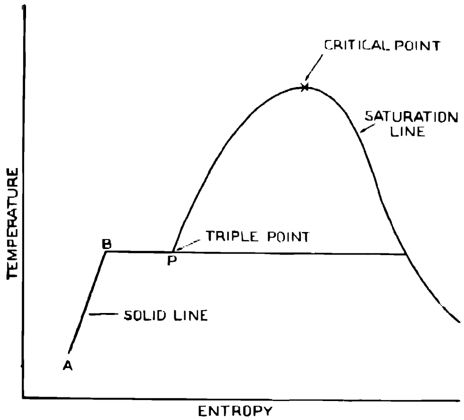

Fig. 3. Temperature-Entropy Diagram

The properties of certain refrigerants, especially carbon dioxide, must be considered in a region which lies outside that already discussed. **Fig. 3** is a *Ts* diagram for carbon dioxide which includes the triple point. At temperatures above that at *P*, the triple point, CO_2 can exist as a liquid, a vapor or a mixture of liquid and vapor, as previously discussed. At the triple point, CO_2 can exist as liquid, vapor and solid, and if either of the two properties, pressure or temperature, is varied, one of the phases will disappear. At any temperature below that at P, CO_2 will exist as a solid to the left of line AB, as a mixture of solid and vapor between the solid and saturation lines, and as a vapor to the right of the saturation line.

First Law of Thermodynamics

9. The **first law of thermodynamics** is a restricted statement of the law of conservation of energy. It is restricted in that it is limited in its application to heat and mechanical energy. The first law of thermodynamics states that heat and mechanical energy are mutually convertible and in a definite quantitative relationship, which is expressed in equation form as $E = JQ$. *E* is mechanical energy, *J*, Joule's mechanical equivalent of heat (778 ft lb per Btu) and *Q* is heat.

It should be understood that the first law does not give an answer to the question of how easily such transformations can be carried out. While any mechanical energy may be converted one hundred per cent into heat, which eventually must necessarily take place, the conversion from heat to mechanical energy is not so easily accomplished. The first law may be extended and written as an energy balance for both static (non-flow) and flow systems. For a static system the energy balance is

$$q = u_2 - u_1 + A \times \text{work}$$

In other words, the law states that any heat which is added to a system increases the internal energy of the system, leaves the system as work, or is divided between the two. If negative signs are given to any of the quantities, a reverse action is indicated, in that heat may flow from the system, the internal energy may decrease, and work may be done on the system.

For a flow system the energy balance is

$$h_1 + AZ_1 + \frac{A\,\text{vel}_1^2}{2g} \pm q \pm A \cdot \text{work} = h_2 + AZ_2 + \frac{A\,\text{vel}_2^2}{2g}$$

h_1	AZ_1	$\frac{A\,\text{vel}_1^2}{2g}$	q	$A \cdot \text{work}$	h_2	AZ_2	$\frac{A\,\text{vel}_2^2}{2g}$
enthalpy	heat equivalent of potential energy	heat equivalent of kinetic energy	heat added or removed	heat equivalent of work added or removed	enthalpy	heat equivalent of potential energy	heat equivalent of kinetic energy

where *Z* represents the static head above some convenient level. The subscripts 1 and 2 refer to any two cross-sections in the path of the fluid. The plus sign before *q* represents heat added and the minus sign represents heat withdrawn from the fluid between the sections 1 and 2, the direction from 1 to 2 being taken as that of the motion of the fluid. The plus and minus signs before work have a similar meaning.

Applying the flow equation to a complete compression system, and assuming no heat leakage, $Q_{evap} - Q_{cond} + A \times \text{work} = 0$, or $Q_{evap} + A \times \text{work} = Q_{cond}$. The heat absorbed by the refrigerant in the evaporator, plus the heat equivalent of the work done by the compressor, is given out to the cooling fluid in the condenser.

The flow equation may be applied to each unit separately. Assuming the static head through each unit to be negligible, the heat leakage small and no appreciable increase in kinetic energy of the fluid through the compressor, the following equations derived from the flow equation are applicable (subscripts refer to Fig. 2):

$\text{work}_{comp} = h_2 - h_1$	compressor
$q_{cond} = -(h_2 - h_3)$	condenser
$h_3 = h_4$	expansion valve
$q_{evap} = h_4 - h_1$	evaporator

10. The use of **tables and charts** may be shown by the solution of a problem.

Consider an ammonia refrigerating system which operates on the ideal cycle shown in Fig. 2. The suction pressure is 20 psia and the discharge pressure 150 psia. The load is 50 tons. Find (a) the weight rate of flow of ammonia, (b) horsepower of the compressor, (c) heat given up to the brine in the condenser, (d) heat absorbed by the refrigerant in the evaporator, and (e) the coefficient of performance.

The values of h for the various places in the cycle may be taken from the Ph chart for ammonia or from the tables. From the Ph chart h_1 is found at the intersection of the line $P = 20$ psia and the saturated vapor line. This is Chart 2 of Chap. 7.

$$h_1 = 606 \text{ Btu}$$

Following an isentropic line (constant entropy) from this point to the intersection of the line $P = 150$ psia, we find

$$h_2 = 734 \text{ Btu}$$

Proceeding from point (2) along the constant pressure line, $P = 150$ psia, to the intersection with the saturated vapor line

$$h_2' = 630 \text{ Btu}$$

The constant pressure line, $P = 150$ psia, is followed further to the left until it intersects the saturated liquid line and we find

$$h_3 = 130 \text{ Btu}$$

Point 4 in the cycle is located by proceeding from (3) along the line of constant enthalpy (vertical on the Ph chart), until intersection with the horizontal line, $P = 20$ psia. Since this change is one of constant enthalpy

$$h_4 = 130 \text{ Btu}$$

(a) weight rate $= \dfrac{200 \times \text{tons}}{(h_1 - h_4)}$ lb per min

$$= \frac{200 \times 50}{(606 - 130)}$$

$$= 21 \text{ lb per min}$$

(b) hp $= \dfrac{21\ (h_2 - h_1)}{42.42}$

$$= \frac{21\ (734 - 606)}{42.42}$$

$$= 63.4$$

(c) q_{cond} = weight rate $(h_2 - h_3)$ Btu per min

$$= 21\ (734 - 130)$$

$$= 12{,}700 \text{ Btu per min}$$

(d) q_{evap} = weight rate $(h_1 - h_4)$ Btu per min

$$= 21\ (606 - 130)$$

$$= 10{,}000 \text{ Btu per min}$$

(e) cop $= \dfrac{\text{heat absorbed in evaporator}}{\text{energy supplied to compressor}}$

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{606 - 130}{734 - 606}$$

$$= 3.72$$

11. Thermodynamic changes from one state point to another may follow any number of paths, depending upon the manner in which the change is carried out.

An **adiabatic** change is one in which no heat flows into or out of the system.

An **isothermal** change is one in which the temperature of the working substance remains constant.

An **isentropic** change requires that the entropy of the substance remain constant in addition to being adiabatic. If the medium is such that it obeys the perfect gas laws in the region in question, the functional relationship between pressure and volume is $Pv^k = \text{const}$, where $k = c_p/c_v$. c_p and c_v are the specific heats at constant pressure and constant volume, respectively.

A **polytropic** change for a gas follows the path represented by $Pv^n = \text{const}$, where $(1 < n < k)$.

12. Work of the compressor is found by use of an expression which is easily developed. **Fig. 4** shows an ideal indicator diagram A-B-C-D for a compressor without clearance. The change A-B is a polytropic compression following the law $Pv^n = c$ from a pressure P_1 and volume V_1 to a pressure P_2 and volume V_2. B-C represents expulsion of the refrigerant vapor at constant pressure P_2 into the high-pressure side of the refrigerating system. D-A represents the suction stroke when the low-pressure refrigerant enters the compressor. Remembering that the area under a curve on the Pv diagram represents work, the area A-B-C-D-A may be seen to represent the work done by the compressor during a cycle.

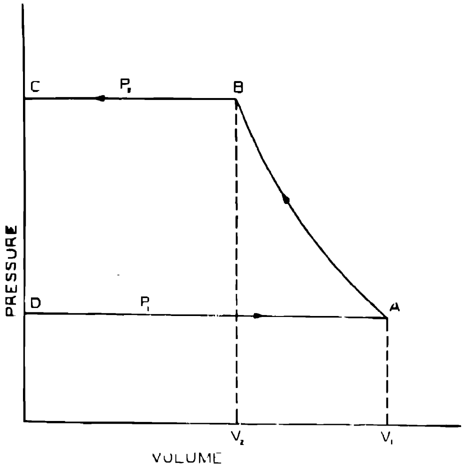

Fig. 4. Ideal Indicator Diagram without Clearance

work during suction stroke	D-A $= P_1V_1$
work during compression	A-B $= \dfrac{(P_1V_1 - P_2V_2)}{(n-1)}$
work during expulsion	B-C $= -P_2V$

The net work for the cycle W_c is found by adding algebraically the separate expressions

$$W_c = \frac{P_1V_1 - P_2V_2}{n-1} + P_1V_1 - P_2V_2$$

$$W_c = \frac{n}{n-1}\,(P_1V_1 - P_2V_2)$$

This equation may be transformed into the form

$$W_c = \frac{n}{n-1}\,P_1V_1\left[1 - \left(\frac{P_2}{P_1}\right)^{(n-1)/n}\right]$$

which shows that the work of compression depends upon the pressure ratio.

The **effect of clearance** may be considered by referring to **Fig. 5**. The changes 3-4 and 5-2 are polytropic. For mechanical reasons it is necessary to have some clearance, which means that some gas volume will remain in the cylinder at the end of the discharge stroke. This high-pressure gas expands behind the cylinder, and the intake valve does not open until the gas pressure drops to that of the intake. The volume V_C is known as the "clearance volume," and the ratio of this volume to the displacement volume, V_C/V_D, is known as the "clearance."

The area 2-3-4-5, which represents the work of the cycle, is the difference between the two areas 1-3-4-6 and 1-2-5-6, each of which represents a no-clearance type cycle. The expression for the work of this type of cycle has been developed.

Subtracting the expressions for the work for these two cycles, work for cycle

$$2\text{-}3\text{-}4\text{-}5 = \frac{n}{n-1}\,P_1V_I\left[1 - \left(\frac{P_2}{P_1}\right)^{(n-1)/n}\right]$$

Thus clearance has no effect on the work, provided the actual volume of gas taken in is used in place of the displacement volume.

13. An expression for the **theoretical volumetric efficiency,** which is the ratio of the theoretical intake volume to the displacement volume, may be easily developed.

$$\eta = \frac{V_I}{V_D} = \frac{V_3 - V_2}{V_3 - V_5}$$

$$V_2 = V_5\left(\frac{P_2}{P_1}\right)^{1/n} \quad \text{polytropic change}$$

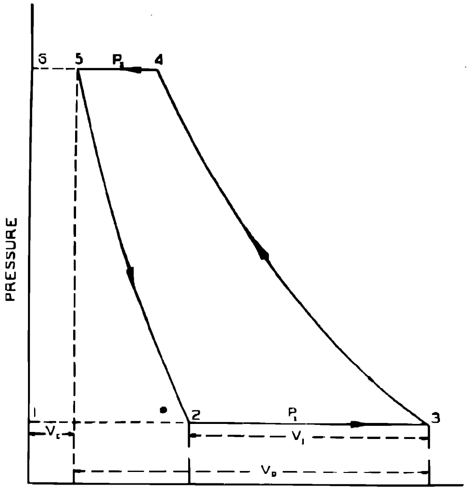

Fig. 5. Ideal Indicator Diagram with Clearance

$$V_3 - V_2 = V_3 - V_5 \left(\frac{P_2}{P_1}\right)^{1/n}$$

we have $\quad \eta = \dfrac{V_3}{V_3 - V_5} - \dfrac{V_5}{V_3 - V_5}\left(\dfrac{P_2}{P_1}\right)^{1/n}$

the clearance $\quad C = \dfrac{V_5}{V_3 - V_5} = \dfrac{V_C}{V_D}$

$$1 + C = \frac{V_3}{V_3 - V_5}$$

thus $\quad \eta = 1 + C - C\left(\dfrac{P_2}{P_1}\right)^{1/n}$

14. The **second law of thermodynamics,** which limits the conversion of heat energy into mechanical energy, may be stated in several different ways. Two statements of the law are: heat of itself cannot flow up a temperature gradient, and no self-acting machine can completely and continuously transform all of the heat supplied to it into mechanical energy.

15. The **Carnot cycle** is a theoretical, highly idealized cycle which permits the determination of the maximum possible amount of mechanical energy which may be derived from a given amount of heat energy. The Carnot cycle on a Ts diagram is shown in **Fig. 6.** The cycle is made up of two isentropic and two isothermal changes. All changes are considered to be reversible. The area under B-C to the *s* axis represents heat supplied to a Carnot engine at a high temperature, T_1, and the area under A-D to the *s* axis represents heat given out by the Carnot engine at the low temperature, T_0. The area A-B-C-D represents the heat equivalent of the work done during the cycle by the Carnot engine.

$$\text{Efficiency} = \frac{\text{output}}{\text{input}} = \frac{Q_1 - Q_0}{Q_1} = \frac{T_1 - T_0}{T_1}$$

Note that the Carnot efficiency increases as T_0 decreases. The practical limit of T_0 is the temperature of the atmosphere. A lower temperature could be maintained, but would require the expenditure of energy as great as that which would be gained by the engine.

A reversible engine working between given temperature limits is the most efficient one which can be operated between the same temperature limits.

Available energy is defined as the maximum conceivable amount of a given amount of heat energy which may be transformed into mechanical energy.

$$Q_{\text{avail}} = Q_{\text{sup}}\left(1 - \frac{T_0}{T_1}\right)$$

The available energy is always less than the energy supplied because it is impossible to expand to absolute zero.

Unavailable energy is the minimum amount of energy which must be rejected from a Carnot engine at the lowest available temperature.

$$Q_{\text{unavail}} = Q_{\text{sup}}\frac{T}{T_1} = T_0 \Delta S$$

This equation states that if Q units of heat are added to a body at a temperature T_1, the unavailable energy of the heat added is given by the product T_0, the lowest available temperature, and the increase in entropy (ΔS) of the body due to the heat added.

If the entropy within a system increases due to friction, the unavailable energy of the system will increase. The increase in unavailable energy accompanying any self-proceeding change in nature is equal to the product of the increase in entropy and the lowest available temperature. Entropy is, therefore, a measure of availability.

Example. 100,000 Btu are supplied to a large body at constant temperature, 212 F.

If the temperature of the atmosphere is 40 F, which may be taken as the lowest available temperature, how many Btu are available for doing mechanical work?

Solution: The increase in entropy of the body due to the heat added is

$$\Delta S = \frac{Q}{T} = \frac{100{,}000}{(460+212)}$$

$$Q_{\text{unavail}} = T_0 \Delta S = 500 \times \frac{100{,}000}{672} = 74{,}405 \text{ Btu}$$

Since the Carnot cycle is a reversible one, heat can be absorbed from the cold body, energy supplied to the Carnot engine from without the system and heat expelled at the high temperature, T_1. The reversed Carnot cycle gives the optimum of performance for any refrigerating machine. It is clear that it would be impossible to attain it in practice.

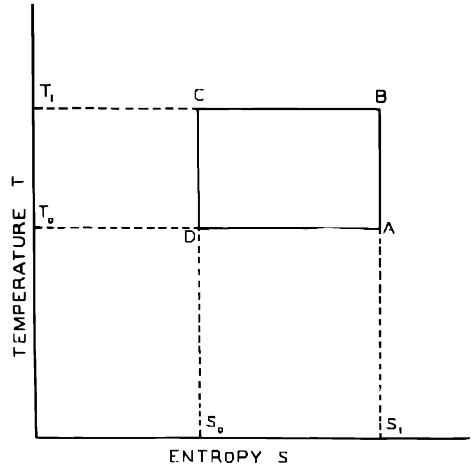

Fig. 6. Carnot Cycle

16. The **coefficient of performance** (cop) of a refrigerating system is defined as

$$\frac{\text{heat absorbed at low temperature}}{\text{heat equivalant of work done}}$$

For the Carnot cycle,

$$\text{cop} = \frac{Q_0}{Q_1 - Q_0} = \frac{T_0}{T_1 - T_0}$$

Example. If 6 hp are supplied to a refrigerating machine operating on the Carnot cycle with a coefficient of performance of 4, (a) how many Btu per min are removed from the cold body; and (b) how many Btu per min are rejected at the high temperature?

Solution: (a) 6 hp is equivalent to 6 × 33,000 = 198,000 ft lb per min

$$Q_1 - Q_0 = \frac{198{,}000}{778} \text{ Btu}$$

$$\text{cop} = \frac{Q_0}{Q_1 - Q_0} = 4$$

$$Q_0 = 4(Q_1 - Q_0) = 4 \times \frac{198{,}000}{778} = 1018 \text{ Btu per min}$$

$$\text{(b)} \quad Q_1 = 5/4 Q_0 = \frac{5 \times 1018}{4} = 1272 \text{ Btu per min}$$

17. One **ton** of refrigeration is defined as 200 Btu absorbed per min or 12,000 Btu per hr. A **ton-day** is equal to 288,000 Btu, which is equivalent to the latent heat of a ton of ice.

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2. COMPRESSION CYCLE OF RECIPROCATING COMPRESSORS

1. The compression cycle starts where the suction vapor enters the suction stop valve and ends where the compressed vapor leaves the discharge stop valve on the way to the condenser. An analysis of the compression cycles of the two most commonly used refrigerants, anhydrous ammonia and Freon-12, is given here.

Since the reciprocating compressor is the most important element of this cycle, its behavior under varying operating conditions will be studied first.

In Chapter 23 the design and handling of industrial compressors are described. It is the purpose here to point out how these two factors affect the operating results of these machines under varying pressure and temperature conditions.

Standard Rating Cycle

2. Some years ago The American Society of Refrigerating Engineers proposed and adopted certain **standard operating conditions** for rating refrigerant compressors. These standard rating conditions are:

Saturated evaporation temperature, 5 F
Suction gas to compressor, 9 deg superheat, 14 F
Saturated condensing temperature, 86 F
Liquid at expansion valve, 9 deg supercooled, 77 F

The corresponding ammonia pressures are 169.2 and 34.27 psia, or a compression ratio of 4.937. For Freon-12 these pressures are 107.9 and 26.51, or a compression ratio of 4.066.

On **Fig. 1** are shown the $T-s$, $P-h$, and indicator card ($P-v$) diagrams of the standard rating cycle, with reference points given in **Table 1** and operating data in **Table 2**. They are based entirely on ideal or theoretical conditions, from which actual machines depart. **Fig. 2** shows the

Table 1. Reference Points for Standard Rating Cycle Analysis

Refrigerant	Point (Fig. 1)	p psia	v cu ft/lb	t F	s Btu/lb F	h Btu/lb
Ammonia	1	169.2	0.02691	86	0.2875	138.9
Freon-12		107.9	0.0124	86	0.05708	27.72
Ammonia	2	169.2	0.02657	77	0.2685	128.4
Freon-12		107.9	0.0122	77	0.05315	25.56
Ammonia	3	34.27	1.176	5	0.2820	128.4
Freon-12		26.51	0.3557	5	0.05589	25.56
Ammonia	4	34.27	8.15	5	1.3253	613.3
Freon-12		26.51	1.485	5	0.17052	78.79
Ammonia	5	34.27	8.241	14	1.3369	618.7
Freon-12		26.51	1.516	14	0.17317	80.04
Ammonia	6	169.2	2.387	222.8	1.3369	720.6
Freon-12		107.9	0.416	109.5	0.17317	91.13
Ammonia	7	169.2	1.772	86	1.1904	631.5
Freon-12		107.9	0.389	86	0.16640	87.37

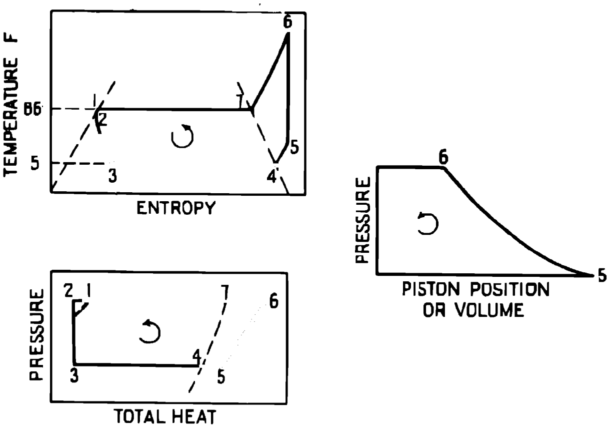


Fig. 1. Rating Cycle (see Table 1)

cfm of ammonia or Freon-12 per ton for various other operating conditions. The three discharge pressures plotted are nearly the same in temperature for the two refrigerants, so that the curves give a close indication of comparative compressor sizes. **Fig. 3** gives adiabatic discharge temperatures which are considerably lower for Freon-12 than for ammonia.

On **Fig. 4** the theoretical adiabatic power requirements for anhydrous ammonia compression at various suction pressures are plotted for three different condenser pressures. The solid curves give the values for single-stage compression and the broken curves for compound compression. The theoretical equivalent values for Freon-12 are about 98% of those for ammonia.

Theoretical Compression Cycle

3. For a clear understanding of the theoretical cycle three diagrams are useful: first, the indicator diagram, **Fig. 5**; second, the temperature-entropy diagram, **Fig. 6**; and third, a plot of temperature against crank position, which is called the time-temperature diagram, **Fig. 7**. The cycle diagram, **Fig. 8**, shows the essential elements of the single-stage compression refrigeration system with lettering to correspond to the points in Fig. 6. Figs. 5 and 7 refer to the compressor alone and Figs. 6 and 8 to the entire system. Compression cycle points are denoted in capital letters and system points in small letters.

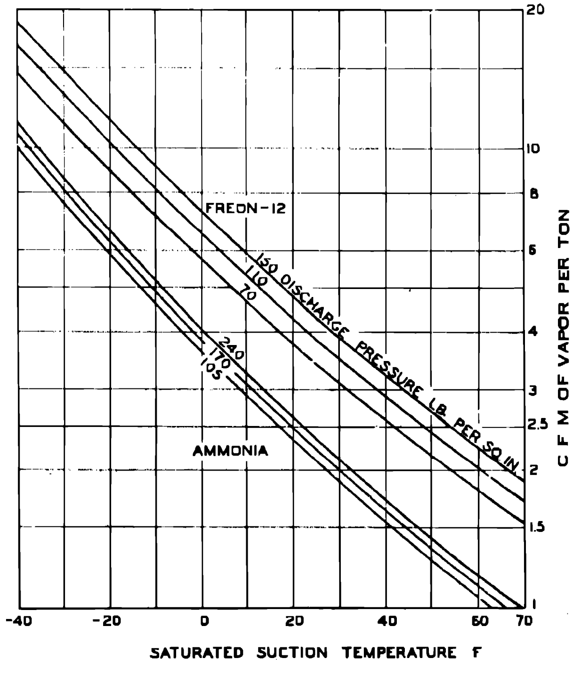


Fig. 2. Cfm Freon-12 or Ammonia per Ton

(Saturated suction and liquid expansion at saturation temperature corresponding to discharge pressure.)

Table 2. Standard Rating Cycle Data

	Ammonia	Freon-12
Refrigerating effect, Btu per lb	490.3	54.48
Cfm per ton	3.36	5.56
Adiabatic hp per ton	0.979	0.959

In Figs. 5 and 7, AB represents the compression, BC the discharge, CD the re-expansion, DA the suction. In Figs. 6 and 8, *a* is the suction from D to A (Figs. 5 and 7), *b* is the discharge from B to C, *c* is the liquid leaving the condenser, e the mixture of liquid and vapor leaving the expansion valve. After passing through the evaporator the cycle is completed at the suction *a*. The line CD is not shown in Figs. 6 and 8. It represents the re-expansion of the vapor trapped in the clearance space of the compressor. Area *a-b-c-d-a* represents the work of adiabatic compression from *a* to *b*, and area *e-f-g-a-e* the refrigerating effect, if distance *e-f*, which is foreshortened in the figure, represents the absolute temperature of the evaporating liquid. In Fig. 7 level DA represents the temperature of the incoming suction gas, and level BC the theoretical discharge temperature, which is also represented by point b of Fig. 6 and developed further in Fig. 3.

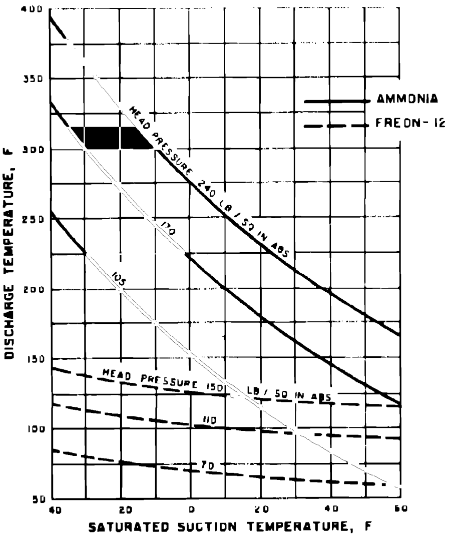


Fig. 3. Adiabatic Discharge Temperatures

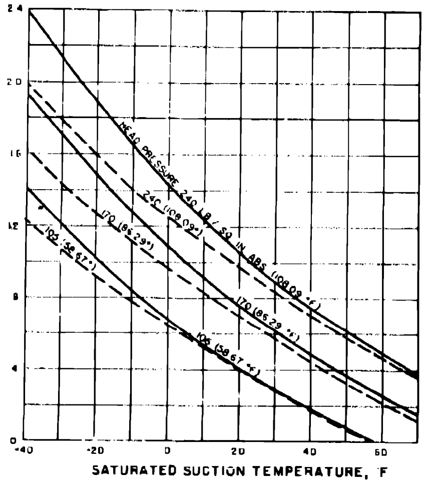


Fig. 4. Theoretical hp/ton for Ammonia Compressor

Values for Freon-12 approximately 98% for same suction and condensing temperatures; broken line curves apply to compound compression with $P_{int} = 1.1 \sqrt{P_{suct} \times P_{disch}}$ 1st stage; discharge water cooled to 108 F and cooled by ammonia to saturation temperature. Liquid cooled by ammonia to saturated intermediate temperature.

Figs. 5, 6, and 7 are constructed for **adiabatic compression**, since it is customary to use adiabatic compression as the standard of comparison on refrigeration duty. Air compressors are frequently com-

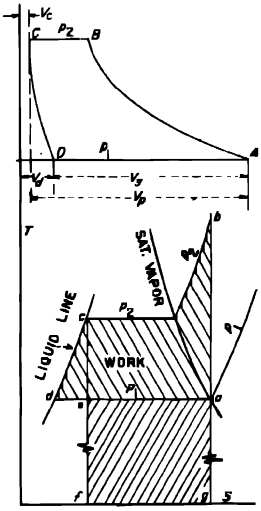


Fig. 5 (above). Indicator Diagram
Fig. 6. Temperature-Entropy Diagram

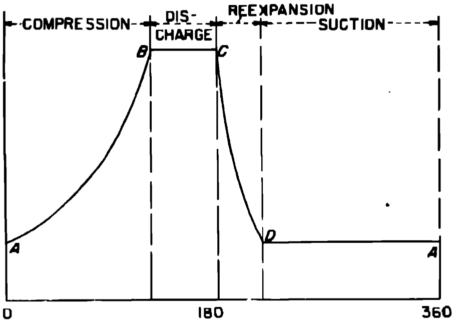


Fig. 7. Theoretical Time-Temperature Diagram

pared with the isothermal, and on indicator cards from refrigerating machines the isothermal compression curve (compression exponent of 1) is frequently drawn in for the purpose of studying valve action in the compressor. Actually, in refrigeration, the isothermal curve has no physical meaning (unless there is extremely high suction superheat), since isothermal compression would result in the discharge of subcooled liquid refrigerant instead of vapor and, due to the change of state, the volume would be very much reduced below the curve with the exponent 1. Furthermore, if a cooling fluid were available at a temperature low enough to equal the suction temperature (again excepting the case of high suction superheat), there would be no need for the refrigerating plant, as the cooling fluid itself could be used directly.

Clearance Volume

4. A reciprocating compressor cannot be designed so that all the vapor contained in it is forced out by the piston at the end of the discharge stroke. There must be some clearance between the piston and cylinder heads at that point of the stroke. The design and location of the compressor valves often leave pockets that the piston cannot fill completely. This total space is called the clearance volume.

The vapor confined in the clearance volume has been compressed to the condenser pressure, and when the suction stroke begins this vapor will re-expand and will not allow the suction valves to open and admit additional vapor until its pressure has been reduced to the suction pressure (Fig. 5). The following is a discussion of the effect the clearance volume has on the theoretical vapor pumping capacity of the compressor.

Nomenclature and Equations (see **Fig.** 5)

V_p = piston displacement
V_c = clearance volume
V_d = re-expansion volume
$V_g = V_p + V_c - V_d$

$$m = \text{percent clearance} = \frac{100V_c}{V_p}$$

n = compression exponent
$V_d = V_c(P_2/P_1)^{1/n}$
$V_g = V_p - V_c[(P_2/P_1)^{1/n} - 1]$
$V_c = mV_p/100$
R = compression ratio
E_1 = volumetric efficiency of theoretical cycle, which is

$$E_1 = \frac{V_g}{V_p} = \frac{V_p - \frac{mV_p}{100}\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right]}{V_p}$$

$$= 1 - \frac{m}{100}\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right]$$

$$= 1 - \frac{m}{100}(R^{1/n} - 1)$$

Tables 3 and **4** respectively list the values of $(R^{1/n} - 1)/100$ for ammonia and for Freon-12. Then the **theoretical volumetric efficiency** = 1 − (percentage of clearance × tabular value).

Applying the clearance factors to the standard rating cycle, the theoretical volumetric efficiency of the compressors operating under these conditions for varying clearance volumes for ammonia is

$$1 - \frac{m}{100}\left[\left(\frac{169.2}{34.27}\right)^{1/1.3} - 1\right]$$
$$= 1 - 0.02415m$$

Theoretical volumetric efficiency of Freon-12 compressor is

$$1 - \frac{m}{100}\left[\left(\frac{107.9}{26.51}\right)^{1/1.138} - 1\right]$$
$$= 1 - 0.02433m$$

These equations show conclusively that for the same percentage of clearance its effect on the theoretical displacement is practically the same for both ammonia and Freon-12. As a matter of fact, the theoretical volumetric efficiency of compressors having the same clearance volume is less for the Freon-12 than for the ammonia compressor, when operated at their respective compression ratios corresponding to the same heat levels for suction and discharge.

For the analysis of the actual compression cycle, the effect clearance has on the weight of refrigerant handled by the compressor in the theoretical cycle must be known.

Referring again to the indicator diagram shown in Fig. 5 it appears that, if:

w_c = weight of the recirculated gas, in lb per min
w_s = weight of suction gas, in lb per min
V_s = specific volume of the suction gas in cu ft per lb,

then

$$w_c = \frac{V_d}{V_s}\,; \qquad V_d = V_c R^{1/n} = \frac{m}{100} V_p R^{1/n}$$

$$w_c = \frac{\frac{m}{100} V_p R^{1/n}}{V_s} \qquad V_p = \frac{V_g}{E_1} = \frac{w_s V_s}{E_1}$$

$$w_c = \frac{\frac{m}{100} R^{1/n} w_s}{E_1}$$

This shows that both the weight of the refrigerant flowing through the compressor and that which recirculates continuously can be calculated for the theoretical cycle. Tables 3 and 4 can be used to obtain $R^{1/n}$ by adding 0.01 to the values given in them for the R under consideration.

The effect of clearance on the theoretical cycle when the ratio of compression is changed is illustrated in **Fig. 9**. In this figure, the theoretical cards of two machines having different clearances are shown. Card AB_1C_1DA is the record of the machine having the larger clearance, V_{C1}; and AB_2C_2DA is the card of the machine having the smaller clearance, V_{C2}, when operating between pressures p_1 and p_2. These machines both handle the same volume of suction gas, V_G, but the larger-clearance machine must have a larger displacement, V_{p1}, than the smaller-clearance machine whose displacement is V_{p2}. The theoretical power consumption of both compressors is the same when operating under the same suction and discharge pressures, p_1 and p_2.

If, now, the suction pressure is raised to p_1', without changing the discharge pressure, the card for machine 1 becomes $A'B_1'C_1D_1'A'$, and of machine 2 becomes $A'B_2'C_2D_2'A'$. In this case, the two machines which formerly had the same capacity, V_G, now have different capacities, V_{G1} and V_{G2}, the larger clearance machine having increased in capacity more than the smaller-clearance machine. Conversely, if the ratio of compression had been increased, the capacity of machine 1 would have decreased more than that of machine 2. On the other hand, of two compressors having the same piston displacement the one having the smaller volumetric clearance will have the larger capacity irrespective of the compression ratio under which they are compared.

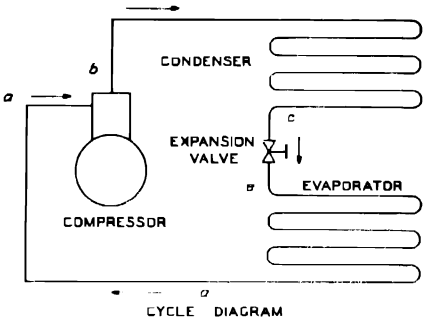


Fig. 8. System Related to Cycle Diagram

These conclusions are true for the theoretical compression cycles of all refriger-

Table 3. Clearance Factors for Ammonia, $(R^{1/n}-1)/100$ when $n=1.3$

R	.0	.2	.4	.6	.8
1	.0000	.0015	.0030	.0044	.0057
2	.0070	.0083	.0096	.0109	.0121
3	.0133	.0145	.0156	.0168	.0179
4	.0190	.0201	.0213	.0223	.0234
5	.0245	.0255	.0266	.0276	.0287
6	.0296	.0307	.0317	.0327	.0337
7	.0347	.0357	.0366	.0376	.0386
8	.0395	.0405	.0414	.0423	.0423
9	.0442	.0451	.0460	.0470	.0478
10	.0488	—	—	—	—

Table 4. Clearance Factors for Freon-12, $(R^{1/n}-1)/100$ when $n = 1.138$

R	.0	.2	.4	.6	.8
1	.0000	.0018	.0034	.0051	.0068
2	.0084	.0100	.0116	.0132	.0147
3	.0163	.0178	.0193	.0208	.0223
4	.0238	.0253	0268	.0282	.0297
5	.0312	.0326	.0340	.0355	.0371
6	.0383	.0398	.0411	.0425	.0440
7	.0456	.0468	.0480	.0495	.0510
8	.0523	.0535	.0549	.0562	.0576
9	.0589	.0603	.0617	.0633	.0645
10	.0657	—	—	—	—

ants. The analysis of the actual compression cycle appearing in this chapter, under the heading "Real Volumetric Efficiency" (page 26), proves that these conclusions are not true for those refrigerants having a high specific weight, such as for instance Freon-12.

Compressor Performance

5. The performance of a compressor departs to a marked degree from the theoretical. To determine the required piston displacement of a compressor or check the size of a compressor for a given refrigerating duty under given temperature and pressure conditions, it is necessary to evaluate as closely as possible the effect which the design of the compressor and the materials from which it is constructed have on the refrigerant gas flowing through it.

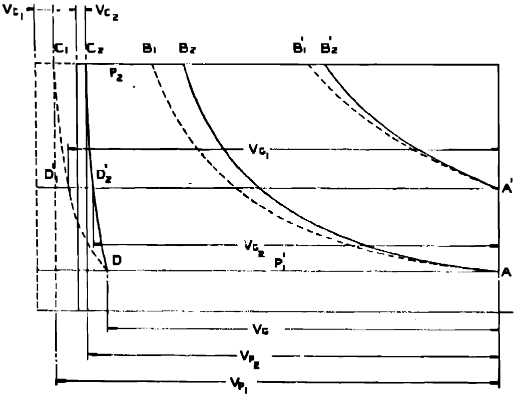


Fig. 9. Effect of Clearance

These two practical limitations demand that the piston displacement of the actual compressor be greater than the volume of the suction vapor drawn to the suction stop valve. The ratio of the volume of the suction vapor and the corresponding piston displacement is called the **total volumetric efficiency** (also real volumetric efficiency and charge volumetric efficiency).

Since the real volumetric efficiency of the actual compression cycle is always less than that of the theoretical cycle, it follows that the power consumption per unit weight of gas handled of the former is greater than that of the latter.

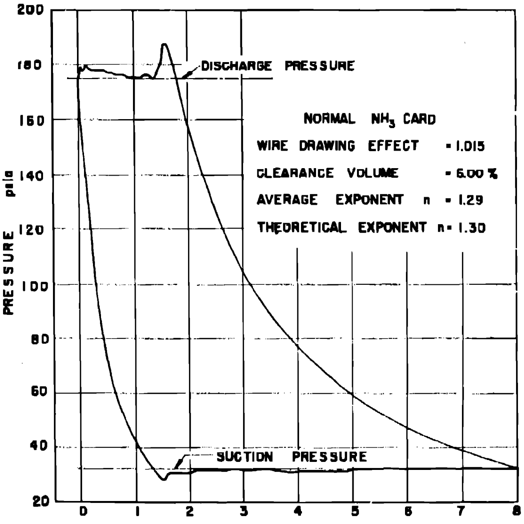


Fig. 10. Normal NH_3 Card

These departures from theoretical calculations are caused by the following factors:

A. Wiredrawing
B. Cylinder heating
C. Suction superheat
D. Water jacketing of compressor cylinders or cooling by radiation
E. Wet compression
F. Speed of rotation.

A. Wiredrawing. A compressor is expected to produce its rated capacity at specified conditions, usually measured at its suction and

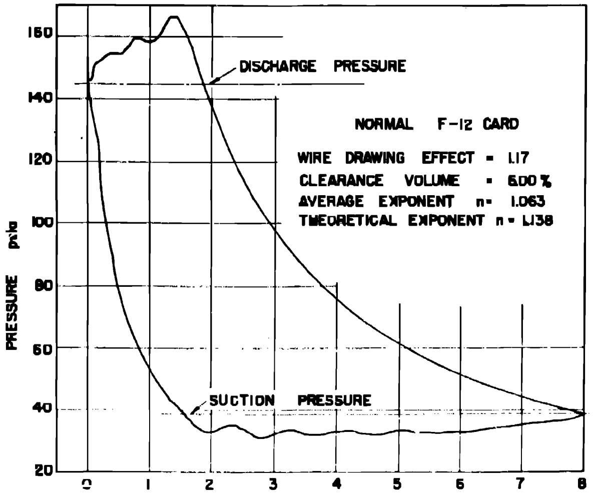


Fig. 11. Normal F-12 Card

discharge stop valves. For flow to take place, there must be pressure losses through the stop valves, compressor passages and compressor valves, including eddy currents within the gas. There must also be sufficient pressure differences across the valves to overcome spring and valve weight and inertia. The power expended on the gas to overcome these losses, occasioned by the design of the compressor, is called wiredrawing. It has the effect on the card (Fig. 5) of bowing the suction line down and the discharge line up, and increasing the indicated work. At the end of each stroke the piston velocity becomes zero, so the lines tend to return to points A and C. The wiredrawing losses add to the heating of the gas during its flow through the compressor.

Figs. 10 and 11 give samples of normal compressor indicator cards of well-designed ammonia and Freon-12 compressors, respectively. The wiredrawing effect is the work equivalent of the area of the indicator card above the discharge pressure line plus the area below the suction pressure line. This effect is generally expressed in terms of the ratio of the total indicator card area and that part lying between the actual suction and discharge pressure lines.

This ratio for the card of the ammonia compressor (Fig. 10) is 1.015, and of the Freon-12 compressor (Fig. 11), 1.17.

Since both these indicator cards are labeled "normal," an explanation of the great difference between the wiredrawing effects of the two must be given.

While the resistance offered to the flow of the refrigerant through the suction and discharge ports and valve housings adds to the wiredrawing, a considerable percentage of this work in a modern well-designed compressor is used up for accelerating the gas through the suction and discharge valves.

In a compressor so designed that the suction gas passes through suction valves in the piston, any initial velocity this gas may have had in the suction main is arrested and turned into heat before the suction stroke begins. The same is true of the gas before it flows through the discharge valves. Therefore the gas must be accelerated from zero velocity. In addition, basing the design of the valves entirely on the ratio of the piston displacement and the effective area of the valve opening is misleading, because the tension of the valve springs decreases this area, particularly at the lower speeds of the piston travel. Therefore a careful study of the loading of the valves, both from the point of view of their correct functioning and also from its effect on the work of wiredrawing, must be made to obtain all-round satisfactory compressor performance.

Again referring to the standard rating cycle, the data in Table 2 show that the weight of refrigerant circulated becomes 0.4079 and 3.671 lb per min per ton for

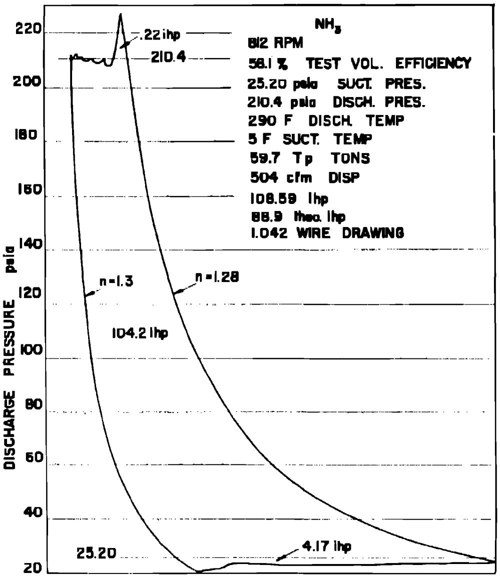


Fig. 12. Wiredrawing Effect—Normal NH$_3$ Card

ammonia and Freon-12, respectively. In this case about nine times as much Freon-12 by weight must be circulated. If the acceleration of the Freon-12 gas passing through the suction and discharge valves is the same as for the ammonia compressor, the wiredrawing effect of the former will be nine times that of the latter. Since the work of acceleration is generally more, the wiredrawing effect is further increased.

A comparison of the four indicator diagrams shown in **Figs. 12, 13, 14,** and **15** leads to a further explanation of the apparent excessive wiredrawing recorded on indicator cards of Freon-12 compressors.

All the indicator diagrams record the pressure and volume changes occurring during the compression cycle in a compressor of the same size operating at somewhat varying speeds and differing pressure ratios. Figs. 12 and 13 show these changes for ammonia gas and Figs. 14 and 15 for Freon-12 for high and low compression ratios, respectively. In **Table 5** appear some data obtained from the analysis of these diagrams.

The high wiredrawing effect shown on cards of Figs. 13, 14, and 15 cannot be accounted for by velocity changes alone. Furthermore, the apparent compression exponents for these three cards is too low. For Freon-12 it is practically isothermal. In addition, the theoretical ihp for the Freon-12 cards is in excess of the actual ihp between the suction and discharge pressure lines. The conclusion must be that

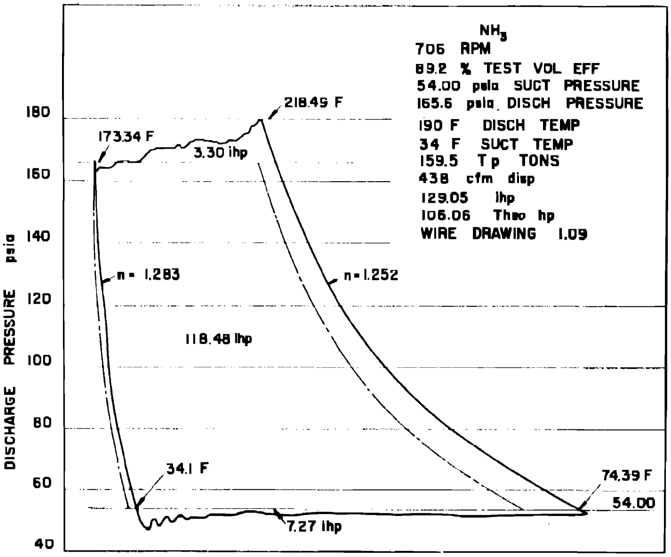


Fig. 13. Wiredrawing Effect—Normal NH$_3$ Card

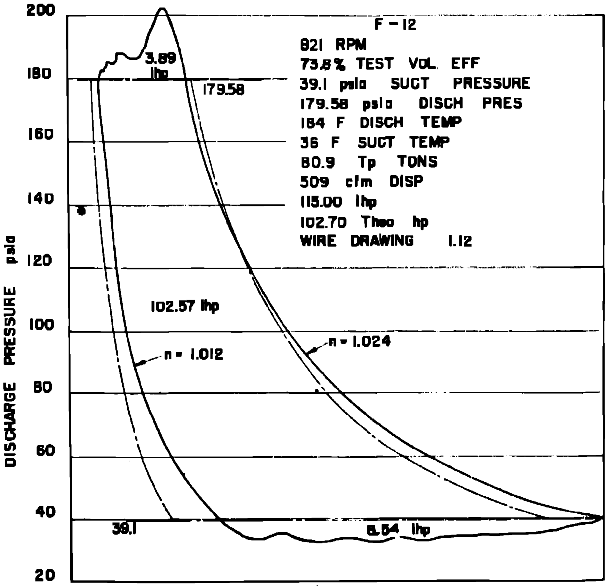


Fig. 14. Wiredrawing Effect—Normal F-12 Card

these indicator cards do not show the actual pressures in the compressor cylinder at the respective locations of the piston. Also, that this lagging action must be the result of the time necessary to overcome the inertia of the refrigerant entering and leaving the instrument. This action is particularly noticeable with Freon-12 on account of its high specific weight. The instrument itself may be partly responsible for these distortions.

Due to the fact that the piston comes to rest and reverses its motion at the end of the stroke, the inertia effect of the gas balances out so that the power measured on the indicator card will be reasonably close to the actual power developed in the compressor.

B. Cylinder heating. Another principal cause for deviation between actual and theoretical performance is the exchange of heat between gas and cylinder walls. Since the cylinder walls (including cylinder head and piston head) are in contact with alternately hot and cold gas (Fig. 7), they will tend to take up a mean temperature somewhere between the two extremes, such that the heat transfer to the walls during the time the gas is hotter than the walls will be exactly balanced by the heat transfer to the gas when it is cooler than the walls, plus the heat leakage outward to the atmosphere surrounding the cylinder, or, if jacketed, to the jacket water. The inner wall surface temperature will then fluctuate about this mean temperature during

Table 5. Comparisons of Ammonia and Freon-12 Indicator Diagrams

Fig. No.	12	13	14	15
Refrigerant	NH_3	NH_3	Freon-12	Freon-12
Compression ratio	8.36	3.0667	4.59	2.646
Compression ratio based on Freon-12	6.86	2.68	4.59	2.646
Wiredrawing	1.042	1.09	1.12	1.29
Apparent exponent — *n*				
for compression curve	1.28	1.252	1.024	1.012
for expansion curve	1.30	1.283	1.012	0.998
Actual ihp between pressure lines	104.2	118.5	102.6	107.4
Total ihp	108.6	129.0	115.0	138.4
Theoretical ihp	88.9	106.1	102.7	108.8

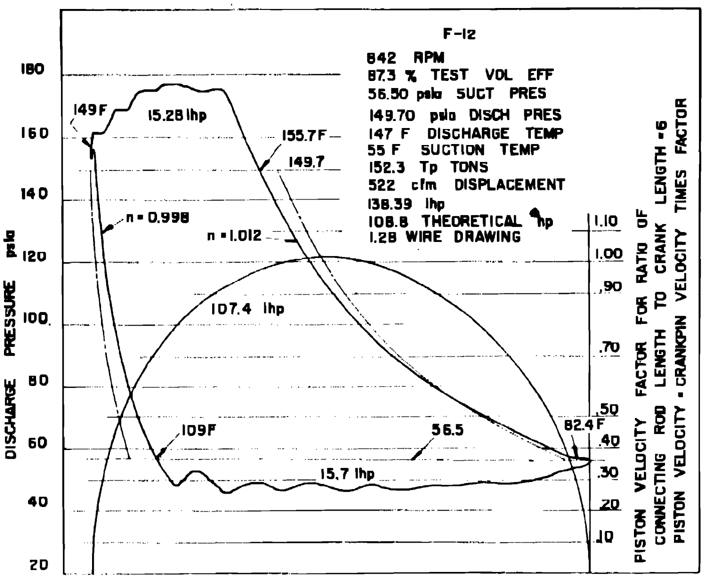


Fig. 15. Wiredrawing Effect—Normal F-12 Card

each revolution by such an amount as is called for by the heat transferred. Various parts of the wall are exposed for various periods during the revolution, so that the mean temperature taken up, and the amount of heat transferred per unit of wall area, will vary with the position of the piston. Wirth has measured the wall temperatures at various points in a compressor.

Fig. 16, which is schematic only, is adapted from Wirth. The curve marked t_g corresponds to Fig. 7 for an actual machine; t_w represents (mean) wall temperature; st_w exposed surface multiplied by mean wall temperature of that surface; and st_g the exposed surface multiplied by gas temperature. The vertical distance between the latter two is then proportional to the heat transfer, which is from the upper curve to the lower one. The dotted line at the beginning of the suction stroke is the temperature of the suction gas. The point where it crosses the dividing line between expansion and suction represents the temperature in the suction pipe. As it mixes with the re-expanded gas at higher temperature, and absorbs heat from the cylinder walls, it rises in temperature.

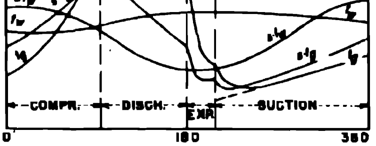
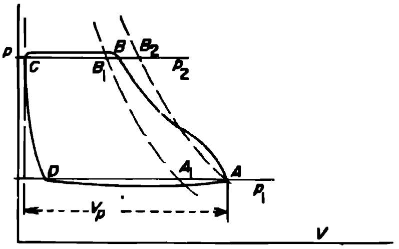


Fig. 16. Time-Temperature Diagram for Actual Case

The theoretical volumetric efficiency is based on having suction gas at the start of the compression stroke at the same temperature as in the suction pipe. The real volumetric efficiency takes into account the temperature of the gas at the end of the suction stroke, and the increased specific volume at that temperature. It has been shown by calculation that the ratio of indicated work to heat absorbed during the suction stroke in a single-stage ammonia compressor varies from about 2.6 to 3.8 with suction pressures between 35 and 5 psig, and with discharge pressures varying from 145 to 205 psig. The discharge pressure makes little difference in this ratio, although much in the actual temperature rise. The heat absorbed per lb of NH_3 varies between 29.7 and 54.6 Btu, corresponding to temperature rises of roughly 55 and 100 F.



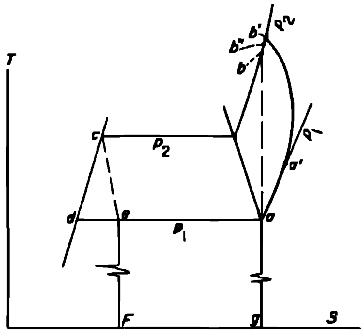


Fig. 17a. Effect of Heat Absorption in Cylinder Walls (P-V)

Fig. 17b. Effect of Heat Absorption in Cylinder Walls (T-S)

During the first part of the compression stroke, more heat is absorbed, but once the gas temperature exceeds the wall temperature, the transfer is reversed, and for the remainder of the compression stroke, the discharge stroke, and the start of the expansion stroke, the gas is giving up heat to the wall. The quantity of heat absorbed by the walls must be equal to the heat previously given up to the gas, plus the heat withdrawn through the walls. Since the time during which this takes place is considerably less than that during which the gas is being heated, the temperature differences must be greater, and the mean temperature of the wall must be closer to the suction stroke gas temperature than to the discharge stroke gas temperature.

Considering the effects on the indicator card and the temperature entropy diagram, we have **Figs. 17a** and **17b.** In Fig. 17a, exaggerated for clarity, DA represents approximately the volume of suction gas at the end of the suction stroke, and DA_1 the volume of the same gas in the suction pipe. The total volumetric efficiency is then DA_1/V_p and the ratio DA_1/DA is the volumetric efficiency due to cylinder heating. Dashed lines A_1B_1 and AB_2 are lines of constant entropy, put in for comparison with the compression line AB. The actual mean discharge temperature is somewhat less than is indicated by point B, since the gas remaining in the cylinder continues giving up heat to the cylinder walls throughout the discharge stroke. The area above line p_2 and below line p_1 represents the wiredrawing loss.

In Fig. 17b, point a represents the condition in the suction pipe, a′ the end of the suction stroke, a′b′ the compression stroke, and b′b″ the discharge stroke, the drop being the cooling by the cylinder walls. The mean temperature of the discharge gas will fall between b′ and b″. Area aa′b′cda does not represent the work of compression but the area e-f-g-a-e, which is fore-shortened, represents the refrigerating effect. If the compression were adiabatic, the excess of *b′* over *b* would represent the increase in work over that of isentropic compression; in the actual non-adiabatic case, *b′* may fall above or below *b*.

During re-expansion, the gas is first cooled by the cylinder walls, and then heated.

C. Suction superheat. Superheat increases the specific volume of the suction vapor. It also increases the total heat of the vapor and hence the refrigerating effect per pound of refrigerant flowing through the cycle, provided the heat absorbed by the vapor in being superheated represents useful refrigeration. Cooling the liquid refrigerant, on its way from the receiver to the evaporator, in countercurrent flow with the suction gas before it reaches the compressor, produces a useful refrigerating effect.

The data presented in **Tables 6** and **7,** also in **Fig. 18,** demonstrate that the refrigerating capacity of Freon-12 compressors increases and the power per ton decreases as the suction gas superheat increases. The reasons for these facts appear clearly from an analysis of the test data in Table 6, the results of which are recorded in the curves shown in **Fig. 19.**

To make this analysis, the equation given in section 4 of this chapter (page 15), showing the relation in the theoretical cycle of the weight of gas, w_s, flowing

Table 6. Test Results on a 15-hp, 1,150 rpm, Completely Enclosed Freon-12 Compressor with a Water-Cooled Motor at 130 psi Discharge Pressure and 37 psi Suction Pressure

(Calculations based on constant volumetric efficiency and constant entropy during compression)

Superheat entering compressor	9.5 F		25.5 F		39.7 F		54 F	
	Test	Calculated	Test	Calculated	Test	Calculated	Test	Calculated
Capacity	100%	100%	104.0	100.5	106.6	101.0	110.0	101.5
Motor input	100%	100%	98.6	100.3	97.5	100.6	96.7	100.9
Total volumetric efficiency	74.5	74.5	77.2	74.5	78.8	74.5	81.0	74.5
Btu per hr per watt	100%	100%	105.5	100.3	109.3	100.5	113.7	100.6
Lb of refrig circulated per min	100%		100		98.6		98.05	

through it and the weight of gas, w_c, recirculated in the cycle, is used.

The equation is:

$$w_c = \frac{\frac{m}{100} R^{1/n} w_s}{E_1}$$

E_1 is the volumetric efficiency of the theoretical cycle. It is equally true of the actual compression cycle if the real volumetric efficiency, E, is substituted for E_1, provided the compressor tested is operated at a speed of rotation which will allow a reasonably accurate functioning of the suction and discharge valves, and has been operated long enough so that the leakage through suction and discharge valves and past the piston is reduced to a minimum consistent with its size and design.

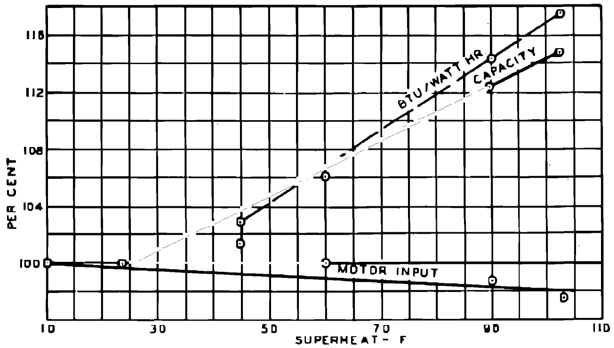
$$E = \frac{v_s}{v_d}$$

where v_s is the volume of suction gas drawn into the compressor and v_d is the compressor displacement.

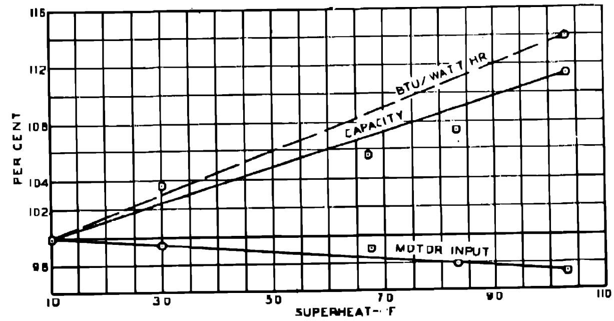
The compressor displacement combines in it the effect that compressor design and operation have on the refrigerant flowing through it.

The equations used in this analysis are therefore the following:

$$w_{c1} = \frac{\frac{m}{100} R^{1/n} w_s}{E_1} = \frac{\frac{m}{100} R^{1/n} v_{d1}}{V_s},$$



36 psi suction pressure
110 psi condensing pressure



14.6 psi suction pressure
110 psi condensing pressure

Fig. 18. Test Results of ¾-hp Air-Cooled Condensing Unit at Different Suction Pressures

used to find the end points of the curves on Fig. 19.

$$w_c = \frac{\frac{m}{100} R^{1/n} w_s}{E} = \frac{\frac{m}{100} R^{1/n} v_d}{V_s}$$

$$V_r = \frac{\left(1 + \frac{m}{100}\right) V_s}{E + \frac{m}{100} R^{1/n}}$$

w_{c1} = the weight of recirculated gas in the theoretical compressor
v_{d1} = the displacement of the theoretical compressor
V_s = the specific volume of the suction gas
V_r = the specific volume of the gas at the end of the suction stroke
m is taken at 6%

Assuming that the capacity of the compressor was 10 tons of refrigeration under the test results and conditions given in column 1 of Table 6, the compressor displacement is found to be 58.38 cfm. The cylinder heating effect for the four tests, when plotted against the volume of suction gas flowing through the compressor in cfm, gives curve I, whose end point is zero cylinder heating for a theoretical volume of suction gas flowing through the compressor of 58.38 × 0.879 = 51.315, where 0.879 = E_1 is the theoretical volumetric efficiency of the compression cycle.

Curve II shows the plot of the real volumetric efficiencies recorded in the test results also plotted against the volume of suction gas. The end point of this curve is at the theoretical volume of 51.315 cfm and $E_1 = 0.879$.

In curve III the suction gas temperature is plotted against the volume of suction gas. The end point of this curve, which shows a suction gas temperature of 142.2 F at a suction gas volume of 51.315 cfm, is found as follows:

The saturated suction gas temperature at a pressure of 37 psia is 21.8 F. The maximum theoretical liquid Freon-12 cooling will be obtained when the liquid is reduced in temperature from the saturation point at the condenser pressure to the saturation point at the suction pressure. This reduction in heat of the liquid must be gained by the suction gas. In this case the corresponding theoretical suction gas temperature is 142.2 F.

Table 7. Test Results on a 20-hp, 600-rpm, Four Cylinder V-Type Open Belted F-12 Compressor with Water Jackets

(124 psi discharge pressure, 37 psi suction pressure)

Superheat	**10 F**	**40 F**	**55 F**	**85 F**
Capacity, %	**100**	**108.5**	**112.1**	**117.7**
Motor input, %	**100**	**101.9**	**101.9**	**100.4**
Btu per hr per watt	**100**	**106.5**	**110**	**117.2**

Refrigerating capacity for the four tests is plotted against suction gas temperature in curve IV. The end point is found at 142.2 F and 12.36 tons of refrigeration. Knowing the thermodynamic properties of Freon-12 at 37 psia and 142.2 F and that the volumetric efficiency at this point is $E_1 = 0.879$, the refrigerating tonnage, 12.36, is readily determined.

Curve V is found by plotting the weight of Freon-12 flowing through the compressor against the suction gas temperature. The end point of this curve is found at a gas temperature of 142.2 F and a weight of 36.5 lb per min. This weight has been calculated, using the above formulas and the end point data found.

The above analysis can be applied to determine whether or not the capacity of an ammonia compressor increases or decreases as the suction gas superheat increases. Having available one test point on the capacity curve, IV, for the ammonia compressor, the end point on it can be calculated.

For example, a 10×10 2-cylinder VSA ammonia compressor, operating at 360 rpm with a suction gas temperature of 15 F, a suction pressure of 20 psig and a discharge pressure of 155 psig, produces a refrigerating effect of 80.6 tons. The theoretical end point for its capacity curve will be reached when the suction gas has a temperature of 168.14 F. The corresponding refrigerating capacity is 79.2 tons. The reduction in capacity from the given test point to the final end point is 1.74%.

For any normal increase in suction gas superheat of an ammonia compressor, its

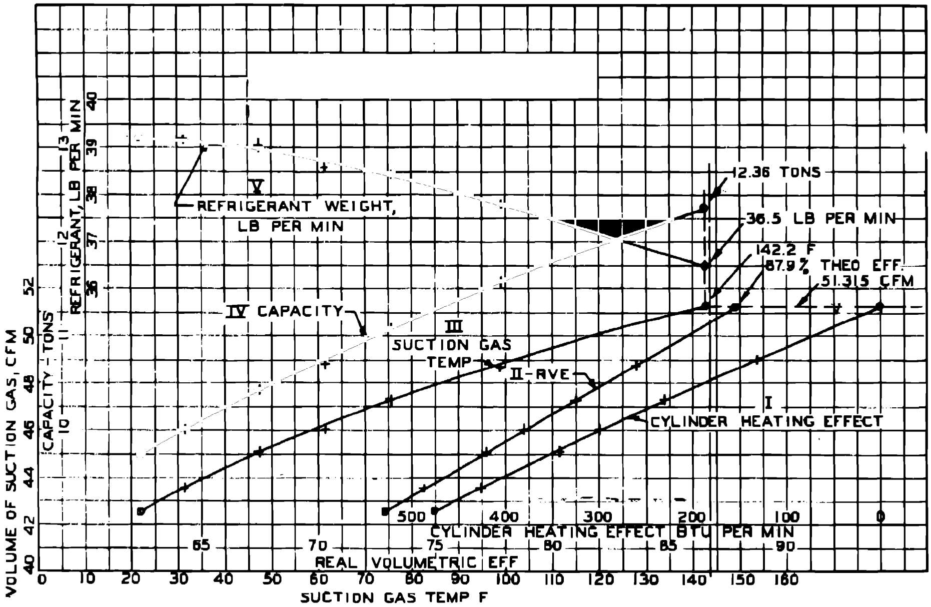


Fig. 19. Effect of Suction Gas Superheat-Analysis of Test Data in Table 6

refrigerating capacity decreases very little as long as the superheat is gained by the suction gas doing useful work such as cooling the liquid ammonia before it reaches the evaporator.

The curves drawn on Fig. 19 establish the fact that when the suction gas superheat increases the cylinder heating effect decreases until it becomes zero at the end point (see curve I). The end point is reached when the volume of suction gas handled becomes equal to the compressor displacement multiplied by the theoretical volumetric efficiency of the compression cycle. At this point the heat added to the suction gas, from its saturation temperature on, is equal to the heat removed from the liquid refrigerant when cooled from its saturation temperature to the saturation temperature of the suction gas.

In practical compressor operation, the maximum useful refrigeration obtainable by the exchange of heat between the suction gas and the liquid is reached when the temperature of the former is raised to the temperature of the liquid as it flows from the condenser.

The plots in Fig. 18 give the increase in capacity and Btu per whr, also the corresponding decrease in motor input, of a small Freon-12 compressor when operated with increasing suction superheat.

D. Water jacketing of compressor cylinders. If the temperature of the jacket water is below the mean wall temperature, the effect of jacketing is to decrease the wall temperature, and therefore decrease the amount of heating during the suction stroke, and also the temperature of the discharge gas. The total volumetric efficiency is increased, and since the entire line ab′ in Fig. 17b is shifted to the left, the work per pound is decreased. Since the machine handles more gas, the power input is not decreased, although this is not true of the work per pound of refrigerant, or the power per ton. Since the wall temperature varies with location, those parts having the highest temperature are most benefited. Obviously, it is not beneficial to jacket walls whose temperature is as low as, or lower than, that of the jacket water. This is reflected in single-acting practice, where the suction valve is in the piston, and only the upper part of the compressor cylinder and the lower part of the discharge chamber are jacketed. Wirth shows that where the mean wall temperature is lower than the condensation temperature corresponding to the discharge pressure, losses similar to those with wet compression occur. Fischer shows improvement with a slow-speed horizontal compressor by jacketing the entire cylinder when compressing superheated vapor, the contrary with wet suction.

Jacketing shows the greatest advantage when high discharge temperatures are encountered, hence ammonia compressors are provided with jackets. Freon-12 compressors are rarely jacketed, usually being provided with radiating fins, or cooling devices are entirely dispensed with, since, due to low adiabatic constant, discharge temperatures are low (Fig. 3).

It is customary to operate ammonia compressors with suction gas at least 10 degrees above the saturated vapor temperature corresponding to the suction pressure. The purpose is to bring to the compressor dry suction gas. Experience indicates that even at this elevated temperature the suction gas is not always dry.

The analysis of the effect of suction gas superheat on the capacity of the ammonia compressor recorded in the preceding section, C, taken together with the heat absorbed by the jacket water, permits the conclusion that even a higher degree of useful suction superheat will not impair the capacity of the compressor. The greater superheat will insure dry suction gas. The higher temperature during the compression cycle will increase the real volumetric efficiency, since the jacket water will remove more of the heat absorbed by the cylinder walls.

E. Wet compression. Theoretically, wet compression is more efficient than dry compression. Referring to Fig. 6, if the line on which points g, a, and b lie were moved to the left until point b were on the saturated vapor line, point a would represent wet suction vapor, and the ratio of the work to refrigerating effect would be a minimum. As compression takes place the contained liquid would be evaporated.

In practice, this is not the case, because of the time element involved in the evaporation. The liquid exists not in molecular suspension but in the form of droplets, and on compression the vapor superheats while the droplets remain in existence. Some evaporation takes place by heat transfer from the gas, but at the end of the discharge stroke a considerable amount of the liquid may remain and continue to evaporate during the expansion stroke. Furthermore, this liquid is now at the higher temperature corresponding to the discharge pressure, and a considerable portion of it evaporates almost instantaneously because the heat necessary is contained in the droplet, and need not be absorbed by heat transfer. The action is similar to flashing on passing through an expansion valve. The expansion stroke with wet compression therefore involves an increase in volume of the refrigerant remaining in the clearance space, reducing the volumetric efficiency without a proportional decrease in power.

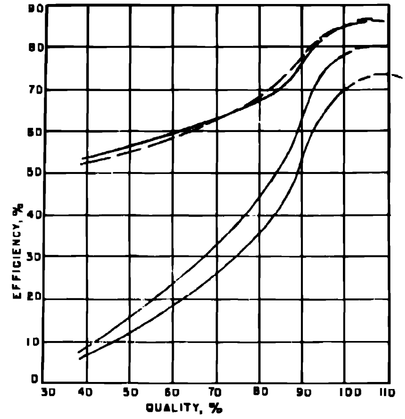


Fig. 20. Effect of Suction Gas Quality on Volumetric Efficiency

Top lines, apparent volumetric efficiency; lower lines, compression efficiency.

Fig. 20 shows the effect of varying the quality of the suction gas, as determined by Fischer.

With ammonia, wet compression is particularly objectionable as the liquid tends to wash oil from the cylinder walls. In enclosed compressors the liquid will run into the crankcase, causing oil foaming and excessive oil pumping.

F. Speed of rotation. For the theoretical compression cycle the refrigerating capacity, the bhp and the volumetric efficiency remain constant per cubic foot of compressor displacement for all speeds of rotation. This is not true for the actual compression cycle.

How these factors vary with speed for a Freon-12 compressor the curves of **Figs. 21** and **22** illustrate. The upper curves show the increase in bhp and refrigerating capacity as the speed increases. The test results are recalculated for a speed of 1,300 rpm and those at 1,300 rpm made equal to 100%. The lower curves show the variation with speed of the bhp per ton and the real volumetric efficiency, respectively.

The bhp and bhp per ton and capacity increase with speed, whereas the real volumetric efficiency increases until the speed of 1,300 rpm is reached and after that decreases.

These variations from the theoretical are brought about mainly by two characteristics inherent in the operation of reciprocating compressors. The one is back leakage of gas through the suction and discharge valves and the piston. This leakage is not proportional to the speed of rotation but depends on the mean effective pressures existing in the compressor cylinder. Its amount increases for the compressor results in Fig. 22 from 300 to 1,700 rpm about 25%, whereas the volume of suction gas handled for the same range of speed increases about 580%.

Considering this characteristic alone the real volumetric efficiency should continue to increase as the speed increases. But the wiredrawing effect interferes. The wiredrawing effect is a function of the volume of gas flowing through the compressor. It is the ratio of the total ihp and the ihp between the actual suction and discharge pressures. When the wiredrawing effect increases the cylinder heating effect also increases.

At the low speed the leakage per cubic foot of gas handled is high. Its effect is to add heat and gas volume during the suction stroke, reducing the volume of suction gas drawn into the compressor. At the same time the effect of wiredrawing on cylinder heating is small, but the action of the sum of the two in reducing the volume of gas drawn in per revolution is greater at that speed than at any intermediate speed up to 1,300 rpm. As the speed of rotation increases, the adverse effect of leakage decreases more rapidly than the increase in cylinder heating due to wiredrawing. At or near 1,300 rpm, the effect of these two adverse conditions is a minimum per revolution and therefore the real volumetric efficiency a maximum.

The real volumetric efficiencies at 1,300 rpm are only 3.9% and about 1% greater than those at 300 and 900 rpm, respectively, for the two tests cited.

Some manufacturers of Freon-12 equipment start the rating of their compressors at this critical speed and limit the maximum speed to about $133\frac{1}{3}\%$ of this speed, as, for instance, for the compressor described in Figs. 21 and 22 from 1,200 to 1,750 rpm. For a larger compressor it may be from 660 to 880 rpm.

The Freon-12 compressor, of which the test results are shown in Figs. 21 and 22, has four cylinders, $2\frac{1}{4}$-in. bore and $1\frac{11}{16}$-in. stroke; cylinder and cylinder heads are finned. The clearance volume is 9% and the maximum operating speed 1,750 rpm.

The curves plotted in **Fig. 23** are obtained as follows:

I—from test results
II—by calculation
III, IV, and V—by use of the weight formulas given in section 5C of this chapter (page 23).

The prolongation of curve III to zero rpm appears to indicate that while maintaining 51.7 psi suction pressure before the suction stop valve and 140.9 psi discharge pressure after the discharge stop valve the total gas leakage will be equivalent to a heat flow of 24 Btu per min at that point.

Real Volumetric Efficiency

6. The effect volumetric clearance, cylinder superheating, leakage, wiredrawing and all other compressor design and operating features have on the value of the real volumetric efficiency varies with type of refrigerant used.

The refrigerants having high specific weights and low enthalpies show the greater variation from the theoretical compression cycle.

Two cases are given here. The first is the relation between the absolute pressure ratio and the real volumetric efficiency and

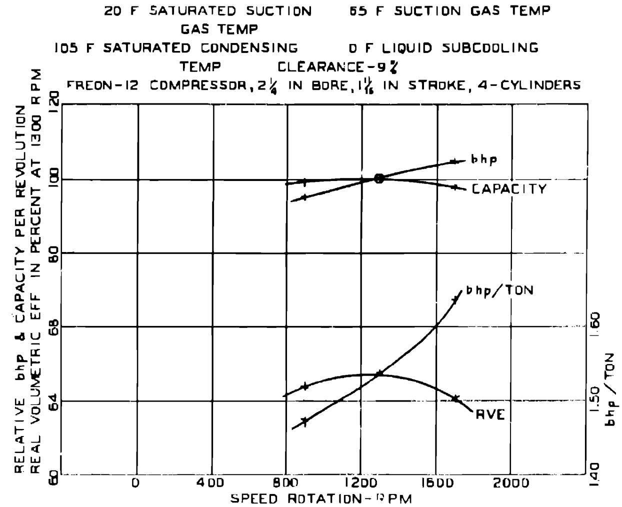


Fig. 21. Effect of Varying Speed of Rotation on Capacity, bhp and RVE

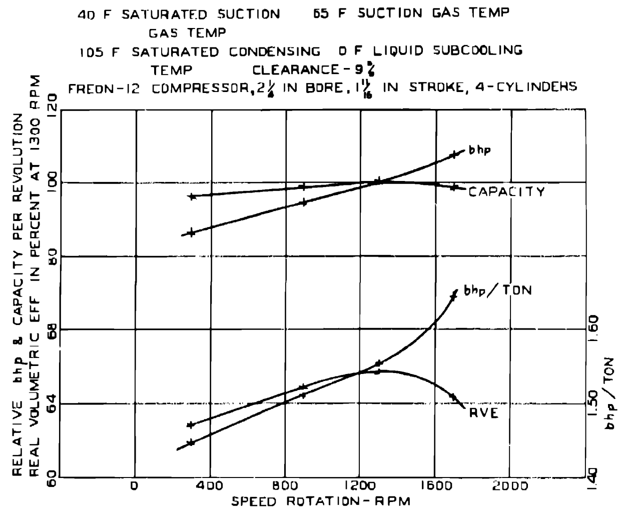


Fig. 22. Effect of Varying Speed of Rotation on Capacity, bhp and RVE

the second the effect of clearance volume on the refrigerating capacity.

(1) **Relation** between the **absolute pressure ratio** and the **real volumetric efficiency** of Freon-12 compressors.

Figs. 24 and **25** present various curves derived from tests on a Freon-12 compressor. The real volumetric efficiency curve plotted on Fig. 24 is obtained by varying the suction and discharge pressures. The test points marked are all within the range of allowable error so that the path of the curve accurately pictures the efficiency for the compressor tested for a considerable variation in these pressures. For comparison the curve of the theoretical volumetric efficiency is added. The lefthand dotted extension, to the line $p_2/p_1 = 1$, indicates that the real volumetric efficiency at this point is approximately 94%.

The curves on Fig. 25 show the real volumetric efficiency and the bhp per ton for the same compressor plotted against the saturated suction temperature.

The data presented on Figs. 21 and 22 and Figs. 24 and 25 were obtained from tests made on the same size Freon-12 compressor. In the former the clearance volume was 9% and in the latter 4½%. The short curves on Fig. 25 show the effect of this difference in clearance volume on the bhp per ton and the real volumetric efficiency at a saturated condensing temperature of 115 F.

As a comparison the curves on **Fig. 26** give values of the real volumetric efficiencies of three two-cylinder, vertical, single-acting ammonia compressors plotted against the absolute pressure ratio. Here

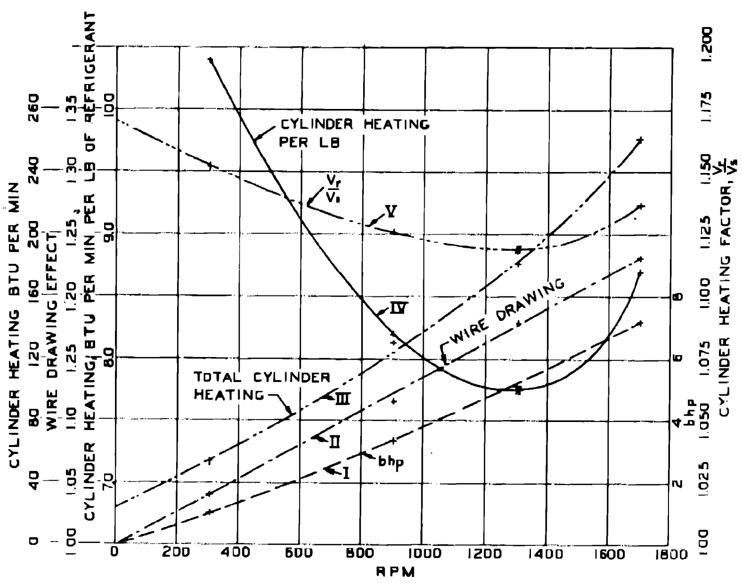


Fig. 23. Analysis of Test Results Shown in Fig. 22

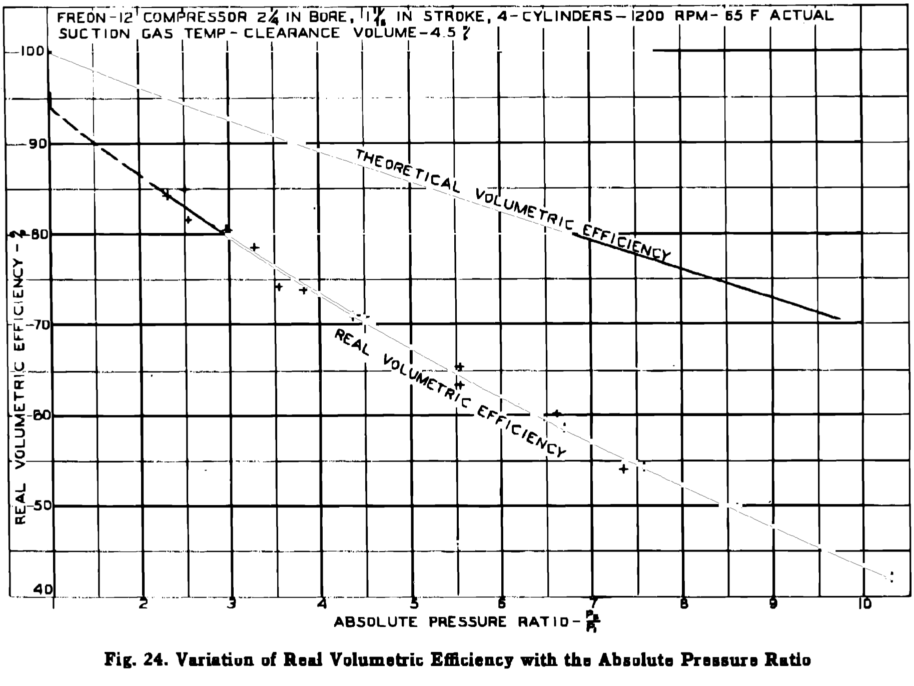


Fig. 24. Variation of Real Volumetric Efficiency with the Absolute Pressure Ratio

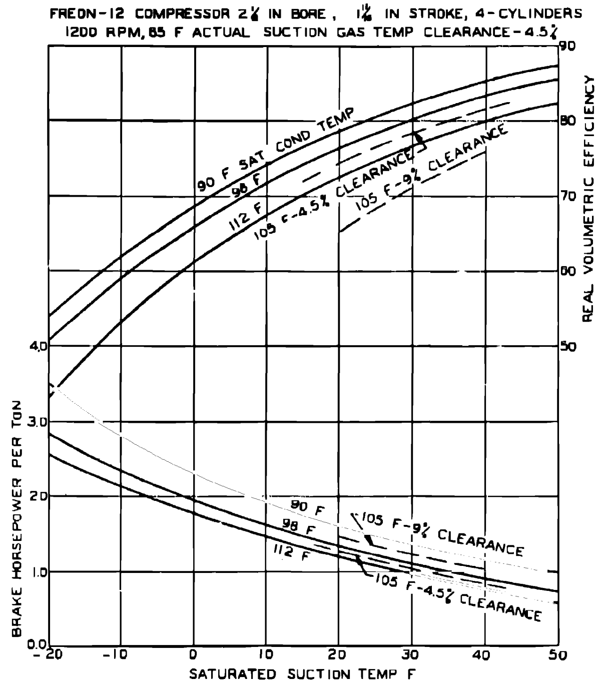


Fig. 25. Variation of Real Volumetric Efficiency with Suction and Discharge Pressures

the test points, particularly for low compression ratios, are located some distance away from a common path drawn through them. For these compressors no single curve will correctly represent the relation between the real volumetric efficiency and the absolute compression ratio for all suction pressures.

The thermodynamic properties of the two refrigerants account for this difference in the behavior of the two gases. As the temperature of the suction gas increases the specific volume of Freon-12 increases less than its enthalpy, whereas for ammonia the reverse is true. As the suction pressure rises the Freon-12 gas can absorb more of the cylinder heating effect than the ammonia gas, due to the increase in wire-drawing for each degree rise in temperature.

(2) The **effect of clearance volume** on the **refrigerating capacity** of Freon-12 compressors.

Fig. 27 presents tests of a 6-in. × 5-in. 4-cylinder Freon-12 compressor operating at 500 rpm. The solid and dashed line curves show the capacity and bhp test results plotted against the saturated suction gas temperature for this compressor with a volumetric clearance of 4.7 and 7.1%, respectively. Tests were run with saturated condenser temperatures of 82, 98 and 112 F.

The two capacity curves of different volumetric clearance intersect at points A, B, and C. An analysis of these test results with the aid of the weight formulas given in section 5C of this chapter (page 23) points to an explanation of this behavior of the compressor. For the theoretical cycle, the greater clearance compressor will always have a lesser refrigerating capacity than the smaller clearance compressor when operated under the same pressure conditions.

The result of the analysis appears in **Fig. 28**. The four curves shown give the change in the cylinder superheat factor,

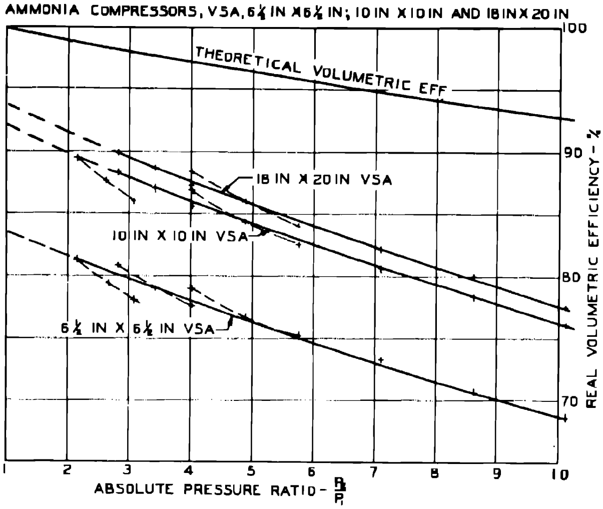


Fig. 26. Variation of Real Volumetric Efficiency with the Absolute Pressure Ratio

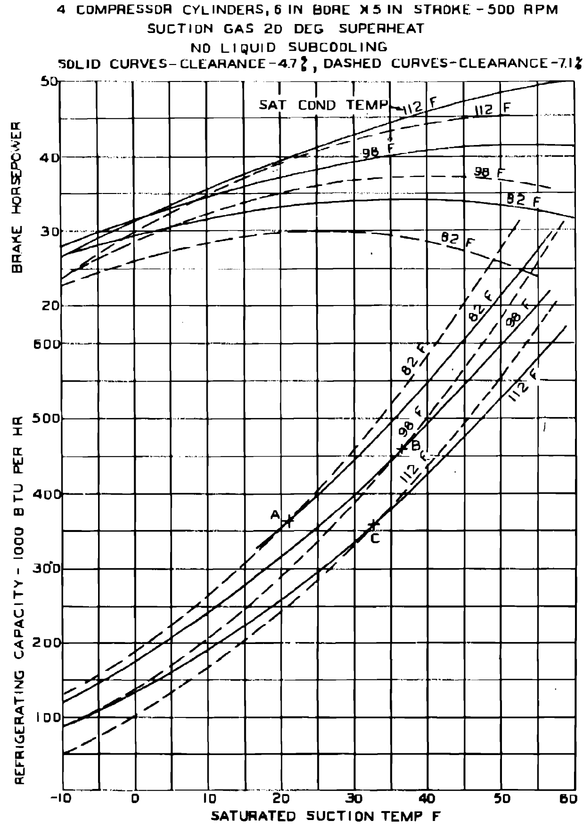


Fig. 27. Effect of Varying Clearance Volume of Freon-12 Compressors on Capacity and Bhp

V_r/V_s, plotted against the saturated suction gas temperature for the 98 F and 112 F saturated condenser temperatures.

The reason for this behavior of Freon-12 compressors is explained by the fact that the weight of recirculated gas plus the weight of the gas discharged by the compressor is greater for the compressor having larger clearance. For the same wiredrawing and somewhat less ihp at the intersecting points, it is evident that the adverse effect of the larger clearance volume is nullified by a lower temperature of the gas at the end of the suction stroke. The result is that at these intersection points the real volumetric efficiency is the same for both. For a further increase in suction pressure, the refrigerating capacity of the larger clearance compressor exceeds that of the smaller clearance compressor. To illustrate, for intersection point B the analysis presents the following results:

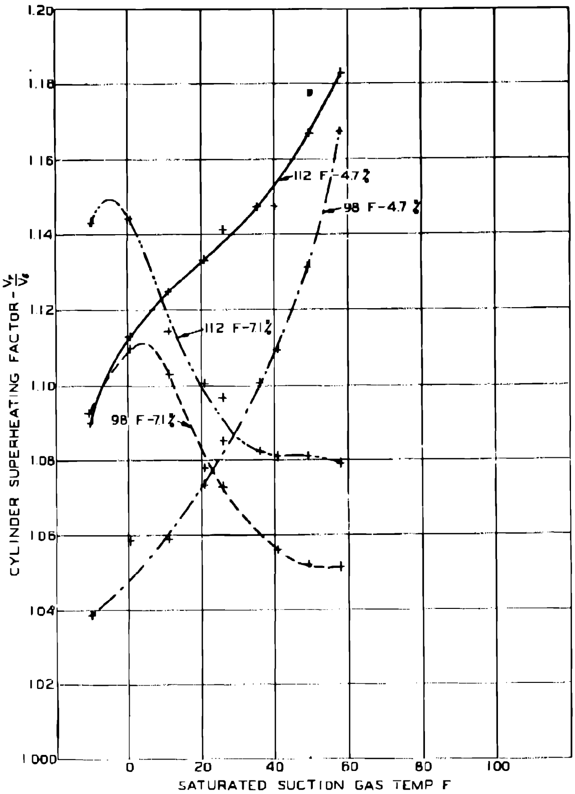


Fig. 28. Effect of Clearance Volume on Cylinder Superheating in Freon-12 Compressors

Clearance volume, %	4.7	7.1
Suction pressure, psia	47.7	47.7
Condenser pressure, psia	128.0	128.0
Refrigerating tonnage	41.75	41.75
Real volumetric efficiency	0.842	0.842
Weight of gas discharged from compressor, w_s, lb per min	153.49	153.49
Weight of recirculated gas, w_c, lb per min	20.36	30.75
$w_c/w_s \times 100$	13.3	20.0
$w_s + w_c$	173.85	184.24
Cylinder heating during suction stroke, Btu per min	1,080	693.5

These calculations affirm that the cylinder heating effect is less for the large-clearance than for the small-clearance compressor in sufficient amount to counteract the greater re-expansion volume of the former.

No single family of curves, based on actual results of tests made on one compressor using any one of the refrigerants, can accurately predict the capacity, volumetric efficiency and other properties of an operating compressor. Effective valve area, clearance, internal cylinder surface in comparison to the gas handled, and many other features of design and operation conspire to make this impossible. There is, however, one basic property of compressors which allows the prediction of capacity, real volumetric efficiency, etc. If a group of compressors varying in size from the smallest to the largest are all based on the same design features, have the same clearance volume, effective valve area, and a somewhat fixed relation of piston diameter to stroke, etc., these predictions are possible. In other words, there is a fixed relation between the various sizes of these properties. To illustrate this fact, the calculations in **Table 8** are presented.

Three sizes in a line of vertical single-acting compressors, ranging from a 3-in. × 3-in. to an 18-in. × 20-in. are chosen. They are the 6½ × 6½, the 10 × 10 and the 18 × 20. Assuming that a complete set of tests at suction pressures ranging in close steps from 0 to 45 psig and condenser pressures ranging from 85 to 245 psig have been run on a 10 × 10 compressor and one test each, say at 20 psig and 155 psig of all the other sizes of the group, complete tables of the properties of all of them can be compiled. Table 8 shows that a constant ratio exists between the real volumetric efficiencies at all suction and discharge pressures between the 18 × 20 and the 10 × 10 compressors, between the 10 × 10 and the 6½ × 6½ compressors, as well as between the 18 × 20 and the 6½ × 6½ compressors. The same is also true of the cylinder heating factor during the suction stroke and the refrigerating capacity.

The same relation may exist between groups of Freon-12 compressors of similar design.

Compound Compression

7. Referring to **Fig. 29**, a-b-c-d is a card for simple compression. In compounding, due to the lower compression ratio, point d would be at d′ and a at a′, for the same gas volume. That is to say, a smaller cylinder can be used on account of higher volumetric efficiency. The card for the low stage then is a′b′c′d′. The low-stage discharge is cooled, reducing its volume to point e. The high-stage card is then e-f-c-g. The area of the compound cards is less than that of the simple card, showing a reduction in indicated horsepower. These cards are theoretical; there are losses in compound compression not found in the simple machine. There are two sets of wire-drawing losses, plus pressure drops through the intercooler and lines, all of which reduce the theoretical savings.

On refrigeration service, compounding offers economies apart from the compression efficiency by cooling liquid refrigerant with the high-stage suction, and by cooling the first-stage discharge, after it leaves the water-cooled intercooler, to a point close to saturation, also by using high-stage suction for direct useful refrigeration. **Fig. 4** shows theoretical power requirements with compound compression. **Fig. 30** is a schematic diagram of a compound system. **Fig. 31** gives tonnage and power of low-stage or booster compressors. These are not test readings, but represent ratings which have

Table 8. Relation between the Real Volumetric Efficiencies and the Cylinder Heating Factors of VSA Ammonia Compressors

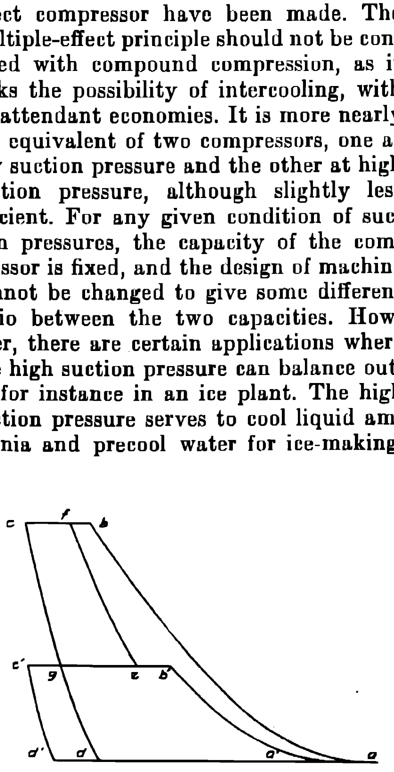
Suction press, psig		5			20			35			50		
Discharge press, psig		125	155	185	125	155	185	125	155	185	125	155	185
RVE	#1—18 × 20	0.820	0.798	0.774	0.883	0.858	0.838	0.900	0.886	0.870			
	#2—10 × 10	0.805	0.782	0.760	0.867	0.842	0.824	0.883	0.870	0.855	0.895	0.875	0.859
	Ratio $\frac{\#1}{\#2}$	1.019	1.021	1.019	1.019	1.019	1.018	1.019	1.019	1.019			
	#3—6½ × 6½	0.732	0.7115	0.686	0.789	0.766	0.751	0.808	0.790	0.776	0.813	0.795	0.780
	Ratio $\frac{\#2}{\#3}$	1.100	1.100	1.109	1.099	1.100	1.098	1.092	1.102	1.102	1.101	1.101	1.101
	Ratio $\frac{\#1}{\#3}$	1.120	1.121	1.128	1.120	1.120	1.116	1.113	1.122	1.122			
$\frac{V_r}{V_s}$	#1—18 × 20	1.13	1.158	1.176	1.096	1.116	1.134	1.09	1.099	1.112			
	#2—10 × 10	1.163	1.179	1.196	1.115	1.139	1.150	1.107	1.118	1.128	1.100	1.107	1.135
	Ratio $\frac{\#1}{\#2}$	0.97	0.983	0.983	0.983	0.98	0.986	0.983	0.982	0.985			
	#3—6½ × 6½	1.270	1.284	1.310	1.219	1.242	1.253	1.206	1.226	1.239	1.208	1.227	1.243
	Ratio $\frac{\#2}{\#3}$	0.915	0.918	0.913	0.915	0.918	0.918	0.918	0.913	0.911	0.911	0.903	0.914
	Ratio $\frac{\#1}{3}$	0.889	0.901	0.898	0.899	0.898	0.906	0.903	0.896	0.897			

proved out in practice. Complete test results on horizontal double-acting compound compressors have been published.

Dual or Multiple-Effect Compression

8. In the dual-effect machine, gas at two suction pressures is handled in the same cylinder. The gas of lower pressure is drawn in on the regular suction stroke, and at the end of the stroke a port is uncovered through which the high-pressure gas is admitted. The theoretical indicator card for such a machine is shown in **Fig. 32** by the solid line. V_a is the volume of low-pressure gas drawn in, V_a' the volume of high-pressure gas, and V_X is the displacement volume lost due to uncovering the port at the end of the stroke, and having to recover it before compression starts.

Mathematical analyses of the multiple-effect compressor have been made. The multiple-effect principle should not be confused with compound compression, as it lacks the possibility of intercooling, with its attendant economies. It is more nearly the equivalent of two compressors, one at low suction pressure and the other at high suction pressure, although slightly less efficient. For any given condition of suction pressures, the capacity of the compressor is fixed, and the design of machine cannot be changed to give some different ratio between the two capacities. However, there are certain applications where the high suction pressure can balance out, as for instance in an ice plant. The high suction pressure serves to cool liquid ammonia and precool water for ice-making.



COMPOUND COMPRESSOR CARD

Fig. 29

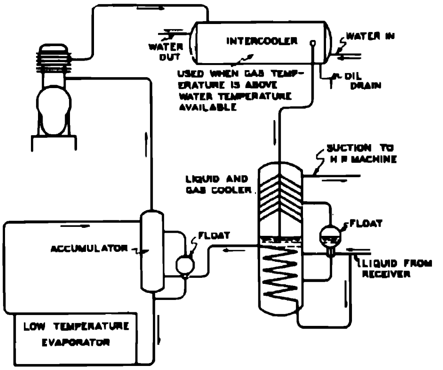


Fig. 30. Schematic Diagram of Compound Compressor

Many ice plants have been equipped with dual-effect machines with considerable reduction in power costs.

If a dual-effect compressor is provided with clearance pockets for capacity control, opening the pockets will reduce the high-pressure capacity and increase the high.

Data comparing a dual-effect CO_2 compressor, cooling the liquid at high suction pressure, with single-stage operation, appear in **Table 9**.

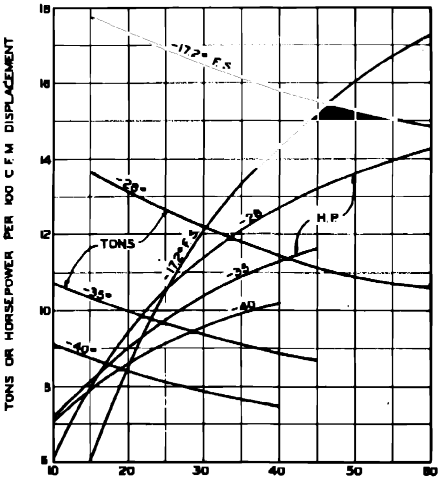


Fig. 31. Horsepower and Capacity Curves for VSA Low-Stage Booster Compressors

[1] Saturated suction hp per 100 cu ft.
[2] Saturated suction tons per 100 cu ft.

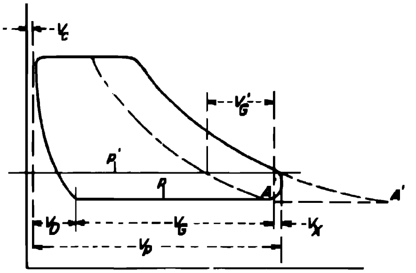


Fig. 32. Theoretical Indicator Diagram

Carbon dioxide compressors are no longer commonly in use for refrigeration but they are used in other departments of refrigerating engineering, especially the manufacture of dry ice. High pressures and low critical conditions are encountered.

Capacity Control

9. **Table 10** shows test results on a 6-cylinder, 25-hp Freon-12 compressor, having suction valves on one or more cylinders held open for capacity control. A comparison of percentage capacity with percentage motor input shows a slight reduction in horsepower per ton, with capacity reduction due to the evaporator temperature increasing and the discharge pressure decreasing, with added steps of capacity reduction.

There may be a considerable difference in the losses in various designs of capacity controls, depending upon gas velocities, viscosity and density of the gas, but a capacity control does always reduce the adiabatic efficiency. There usually is, however, an actual saving in power as compared to operating at full capacity on a greatly reduced load. Furthermore, for proper regulation, some adjustment of machine capacity is essential in certain applications. A change of compressor speed, or providing a number of compressors and operating as many as are needed for the load, would be a more efficient means, but far more expensive to install.

Some of the various capacity control means listed in Chapter 23, while feasible, are not current practice. Adjustable clearance, one of the more popular devices, can be finely adjusted for any capacity within its range, whereas other methods are operative in rather wide steps. The actual reduction with any particular clearance setting varies with the compression ratio.

Performance Data

10. **Tables 11 and 12** (see pages 36 and 37) give performance figures for ammonia compressors and **Table 13** (see page 37) shows the corresponding volumetric and adiabatic efficiencies. The above tables represent a consensus of compressor manufacturers' values rather than actual tests.

Exhaustive data on an ammonia com-

Table 9. Test Results of CO_2 System with Dual Effect

Temp of evaporation, F	Temp of liquid to exp valve, F	Pressure of evaporation, psig	Pressure of condensation, psig	Ratio of compression	Temp of inter. receiver, F	Approx press of intermediate receiver, psig	Ratio of compression, hp to int	Single compr'n theo displ't saturated, cu in./min/ton	Actual single compr displ't, cu in./ton, working cond'n	Dual compr'n displacement, cu in./min/ton, working cond'n	Theo. adiabatic hp/ton single compr'n saturated	Actual bhp/ton aver. work. condition single compr'n	Actual bhp/ton dual compr'n	Theo. adiabatic disch temp F 20 deg superheat single compr'n	Actual disch temp F 20 deg superheat single compr'n	Actual disch temp F 20 deg superheat dual compr'n	Coefficient of performance single compr'n theo adiabatic
−20	85	206	1,130	5.5	19	399	2.83	228	2,960	1,940	2.68	3.2	2.94	213	218	190	1.76
−10	85	247	1,125	4.55	26	445	2.53	1,890	2,440	1,620	2.28	2.75	2.53	206	203	176	2.06
± 0	85	294	1,125	3.83	30	475	2.37	1,570	2,010	1,380	1.88	2.35	2.12	192	192	165	2.50
5	85	320	1,125	3.52	33.5	500	2.26	1,440	1,840	1,270	1.71	2.16	1.96	187	186	161	2.78
10	85	347	1,120	3.24	39.0	540	2.07	1,315	1,685	1,170	1.54	1.97	1.79	181	177	158	3.08
15	85	376	1,120	2.98	41.5	564	1.98	1,215	1,555	1,080	1.41	1.79	1.63	175	171	155	3.36
20	85	407	1,120	2.75	43.5	580	1.93	1,120	1,440	1,010	1.28	1.61	1.48	170	165	151	3.69
25	85	440	1,120	2.54	47.0	610	1.84	1,030	1,310	945	1.16	1.46	1.34	165	160	148	4.09
30	85	475	1,120	2.36	49.5	630	1.78	953	1,230	880	1.06	1.30	1.20	158	156	146	4.48

Table 10. Tests on Freon-12 Compressor, 6-cyl Unit under Varying Load

Number of cylinders operating	Discharge pressure, psi	Evap temp, F	Capacity, 1,000 Btu/hr	Motor input, kw	Capacity, %	Motor, input, %
6	119	38.0	277	22.1	100	100.0
5	117	41.3	248	19.1	89.5	86.5
4	112	44.5	216	16.1	78.2	73.0
3	107	48.7	117	13.3	63.9	60.0
2	102	53.0	119	10.2	43.0	46.0

pressor appear in **Tables 14, 15, 16** (see page 38). The compressor was a 4.5-in. ×4.5-in. single-cylinder machine running at from 444 to 537 rpm, with feather valves, operated by means of a 10 to 25-hp dynamometer-motor, direct-connected through flexible coupling to compressor shaft. The speed was controlled by resistance grids, permitting the brake and indicated horsepower to be measured. Actual capacity was measured by weighing ammonia from a multitube multipass condenser having two shells, each with seven tubes, with a condensing surface of 93.8 sq ft. The evaporator operated on brine, heated in turn by steam. Suction gas temperature at the machine was held at a few degrees of superheat. Liquid at the expansion valve had the same temperature as the liquid leaving the condenser.

Comparison of the actual compression cycle with the theoretical can be made by reference to Table 16, which appears in connection with studies of volumetric efficiency of the same compressor. The author of these tests commented that the power requirements of this compressor seemed high, but made no statement as to whether they were typical of this whole class of machines.

There is only one set of test data on volumetric efficiency in print, this one based on measurement of ammonia flowing from which was determined the total volumetric efficiency under various operating pressures. Knowing the amount of clearance in the existing compressor, the investigators computed volumetric efficiency due to clearance. They then divided the "total volumetric efficiency" by the "clearance volumetric efficiency" and obtained what was termed the real volumetric efficiency, taking account of the loss due to superheating the gas as it entered the cylinder, and also whatever losses may have existed in leakage. It was then shown that real volumetric efficiency, as well as efficiency due to clearance, is a function of the compression ratio alone. **Fig. 33** shows real volumetric efficiency plotted against compression ratio. The lowest curve is an exact reproduction of the curve of real volumetric efficiency in the tests. Confirming test results falling close to this curve were later obtained on a 3-in. × 3-in. compressor and a 3½-in. × 3½-in. compressor.

The engineer strives to develop more and more efficient apparatus, but high efficiencies are not an end in themselves. Reliability and low upkeep cost can outweigh a few points of efficiency, and other factors such as quietness, freedom from vibration, reduced size and weight, may be of primary importance.

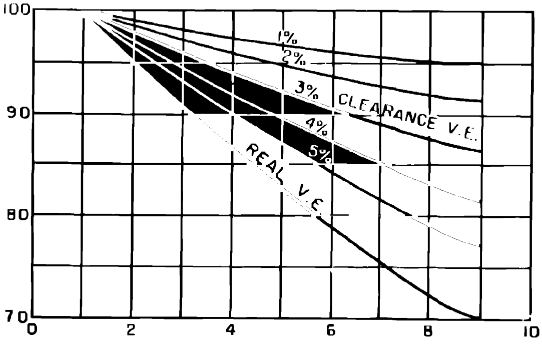


Fig. 33. Real Volumetric Efficiency vs Compression Ratio

Table 11. Ammonia Compressor Performance* at Various Operating Conditions for Vertical Single-Acting Ammonia Compressors and Single-Stage Compression (Air Conditioning and Refrigeration Machinery Association)

Pressure and corresponding saturated temperature of suction gas		Discharge pressure, psig															
		135		145		155		165		175		185		195		205	
		Corresponding saturated temperature, F															
		78.7		82.5		86.2		89.7		93.0		96.2		99.3		102.3	
Pressure, psig	Temperature, F	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$	$\frac{\text{cfm}}{\text{ton}}$	$\frac{\text{bhp}}{\text{ton}}$
0	−28.0	12.34	2.51														
5	−17.2	8.42	2.02	8.64	2.12	8.89	2.22	9.16	2.32	9.40	2.41	9.66	2.50				
10	− 8.4	6.36	1.694	6.53	1.782	6.70	1.869	6.84	1.955	7.01	2.04	7.16	2.12	7.35	2.20	7.53	2.28
15	− 1.0	5.09	1.449	5.21	1.531	5.34	1.611	5.45	1.690	5.56	1.765	5.67	1.839	5.79	1.911	5.93	1.980
20	5.5	4.28	1.270	4.39	1.341	4.46	1.412	4.55	1.482	4.64	1.552	4.72	1.621	4.80	1.688	4.90	1.755
25	11.3	3.69	1.127	3.75	1.193	3.84	1.259	3.91	1.324	3.98	1.387	4.06	1.449	4.12	1.509	4.20	1.568
30	16.6	3.24	1.011	3.31	1.073	3.38	1.133	3.44	1.193	3.50	1.251	3.56	1.307	3.63	1.363	3.69	1.419
35	21.4	2.92	0.910	2.96	0.971	3.00	1.029	3.06	1.085	3.11	1.140	3.16	1.193	3.22	1.244	3.27	1.293
40	25.8	2.62	0.825	2.66	0.883	2.71	0.938	2.75	0.992	2.81	1.044	2.85	1.094	2.90	1.144	2.94	1.193
45	30.0	2.39	0.755	2.43	0.807	2.47	0.859	2.50	0.910	2.54	0.961	2.58	1.009	2.62	1.057	2.65	1.103
50	33.8	2.18	0.689	2.21	0.739	2.25	0.789	2.28	0.838	2.33	0.886	2.36	0.934	2.39	0.980	2.24	1.026

* Based on two-cylinder compressors with 6-in. bore and 6-in. stroke. For other compressor sizes, multiply by machine factors of Table 12.

The values above the heavy line are for ratios of compression above 8.0, and for these conditions it is recommended that multi-stage compression be considered.

$\frac{\text{cfm}}{\text{ton}}$ = average values of swept volume, in cu ft per min, required per ton refrigeration based on a suction superheat of 10 F and on liquid ammonia entering the refrigerant control valve at the saturated temperature corresponding to the discharge pressure.

$\frac{\text{bhp}}{\text{ton}}$ = average values of actual compressor brake horsepower per ton refrigeration.

Table 12. Machine Factors for Various Compressor Sizes (Air Conditioning and Refrigeration Machinery Association)

	Compressor bore and stroke, in.																
	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8	8½	9	9½	10	10½
cfm/ton	1.034	1.030	1.026	1.022	1.017	1.012	1.006	1.000	0.993	0.986	0.979	0.972	0.966	0.960	0.954	0.948	0.943
bhp/ton	1.132	1.106	1.083	1.063	1.044	1.027	1.013	1.000	0.988	0.977	0.968	0.960	0.952	0.944	0.937	0.930	0.924

Table 13. Volumetric Efficiency and Adiabatic Efficiency of Vertical Single-Acting Single-Stage Ammonia Compressors from Table 11

Pressure and corresponding saturated temperatures of suction gas		Discharge pressure, psig															
		135		145		155		165		175		185		195		205	
		Corresponding saturated temperature, F															
		78.7		82.5		86.2		89.7		93.0		96.2		99.3		102.3	
Pressure, psig	Temp, F	Vol eff	Adiab eff	Vol eff	Adiab eff	Vol eff	Adiab eff	Vol eff	Adiab eff	Vol eff	Adiab eff	Vol eff	Adiab eff	Vol eff	Adiab eff	Vol eff	Adiab eff
0	−28.0	62.5	61.0														
5	−17.2	69.1	63.0	68.1	64.2	66.8	64.0	65.4	64.2	64.3	64.0	63.0	64.0				
10	− 8.4	73.6	67.0	72.5	67.0	71.1	67.2	70.4	66.9	69.2	67.0	68.2	67.0	67.1	67.0	65.9	67.0
15	− 1.0	75.8	68.8	75.2	68.8	74.4	69.5	73.5	69.0	72.7	69.0	72.0	69.0	71.2	69.0	69.8	69.0
20	5.5	78.6	70.0	77.2	70.5	76.6	70.5	75.7	70.5	74.9	70.7	74.3	70.7	73.5	71.0	72.6	71.0
25	11.3	80.1	70.7	79.7	71.5	78.6	71.6	77.7	71.7	76.8	72.1	75.7	72.5	75.2	72.3	74.5	72.5
30	16.6	81.2	70.8	80.0	71.4	79.3	72.0	78.5	72.6	77.8	72.9	77.2	73.3	76.3	73.3	75.6	73.8
35	21.4	81.6	72.1	81.3	72.7	80.7	73.0	79.7	73.7	79.1	74.3	78.4	74.5	77.6	74.8	77.3	75.3
40	25.8	82.3	71.7	82.0	72.5	81.3	73.2	80.6	73.8	79.7	74.3	79.3	75.2	78.7	75.2	78.2	75.6
45	30.0	83.2	70.4	82.6	73.0	81.8	72.2	81.5	73.9	81.1	74.5	80.3	75.3	79.4	75.2	79.2	75.5
50	33.8	84.6	70.8	84.2	72.2	83.3	73.2	82.9	74.1	81.8	74.6	81.3	75.3	81.0	75.8	80.6	76.5

Table 14. Capacity of Commercial Ammonia Compressor as a Function of Pressure

(Tabular values are tons refrigerating capacity of machine)

Disch press, psia	Suction pressure, psia			
	50	40	30	20
110	7.6	5.9	4.3	2.6
130	6.9	5.6	4.1	2.4
150	6.2	5.4	3.9	2.3
170	5.4	4.9	3.7	2.1
190		4.8	3.4	1.9
210			3.2	1.8

Table 15. Horsepower per Ton—for $4\frac{1}{2} \times 4\frac{1}{2}$ s.a., single-cyl Ammonia Compressor at 500 rpm

Disch press, psia	Cond temp, F	Suction pressure, psia											
		50 (21.67 F)			40 (11.66 F)			30 (−0.57 F)			20 (−16.64 F)		
		1	2	3	1	2	3	1	2	3	1	2	3
100	56.05	0.74	0.38	0.54	0.96	0.50	0.72	1.23	0.66	0.95	1.77	0.95	1.45
120	66.02	0.86	0.49	0.66	1.11	0.63	0.85	1.41	0.81	1.12	2.03	1.11	1.62
140	74.29	0.98	0.60	0.78	1.26	0.74	0.97	1.60	0.94	1.27	2.29	1.25	1.80
160	82.64	1.11	0.70	0.90	1.42	0.85	1.09	1.78	1.05	1.41	2.56	1.37	1.99
180	89.78	1.23	0.81	1.02	1.56	0.94	1.20	1.96	1.15	1.54	2.82	1.50	2.14
200	96.34	1.36	0.91	1.13	1.72	1.01	1.30	2.14	1.25	1.65	3.08	1.60	2.31
220	104.42	1.48	1.02	1.25	1.89	1.08	1.39	2.32	1.34	1.75	3.35	1.69	2.48

1. Brake hp   2. Indicated hp   3. Indicated+friction hp

Table 16. Volumetric Efficiency of Commercial Ammonia Compressor

(Log of tests, varying pressure of suction gas; same machine as in Tables 14, 15.)

Rpm	540	540	532	530	519
Suct gas, temp, F	51.0	47.0	55.0	56.5	75.2
Disch gas, temp, F	277	244	269	190	263
Disch gas, temp F theo	282	223	246	170	240
Suct press, psia	24.34	34.71	44.07	54.71	63.28
Disch press, psia	133.1	134.6	186.0	137.4	215.2
Weight NH_3 circ, lb/min	1.233	1.977	2.386	3.445	3.395
Total real vol eff	67.6	76.8	74.7	86.7	78.3
Real vol eff without clearance	80.0	85.8	84.6	92.2	86.4

3. ABSORPTION REFRIGERATING SYSTEMS

Absorption refrigerating machines are **heat-operated,** and make use of at least two working substances, a **refrigerant** and an **absorbent** for the refrigerant. **Liquid-absorbent** refrigerating systems operate in accordance with the laws and properties of solutions, particularly with respect to exchange of materials between the liquid and vapor phases. A simple continuous cycle for a liquid absorbent is shown in **Fig. 1.**

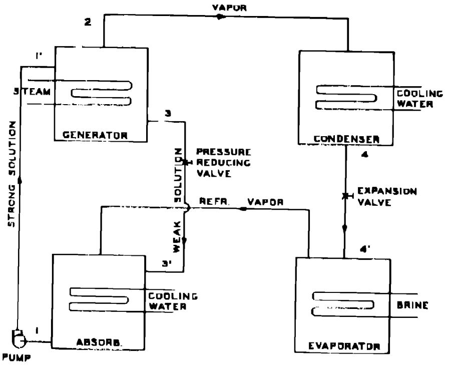


Fig. 1. Simple Absorption Cycle

I. Laws and Properties of Solutions

1. If two or more substances mix uniformly over a range of compositions, to produce only one physically distinct **phase,** such a mixture is called a **solution,** as distinguished from a mechanical mixture of two or more phases. Here, the term "solution" will usually refer to a liquid phase. In describing solutions and gaseous mixtures, the **compositions** may be specified, for instance by giving either the **weight fractions** or the **mol fractions,** or the equivalent percentages, of the pure constituents or components. Or the **concentrations** of the components per unit volume may be specified in any of numerous ways. For a two-component, refrigerant-absorbent system:

Weight fraction:

$$x_1 = \frac{G_1}{G_1 + G_2};\quad x_2 = \frac{G_2}{G_1 + G_2};\quad x_1 + x_2 = 1$$

Mol fraction:

$$N_1 = \frac{G_1/m_1}{G_1/m_1 + G_2/m_2};\quad N_2 = \frac{G_2/m_2}{G_2/m_1 + G_2/m_2}$$

$$N_1 + N_2 = 1$$

where G_1 = weight of refrigerant in mixture, lb
m_1 = molecular weight of refrigerant
G_2 = weight of absorbent in mixture, lb
m_2 = molecular weight of absorbent

Where no confusion will result, x and N may be used without subscript to refer to the refrigerant. A **weak** solution is one which contains relatively little refrigerant, as compared with a **rich** or **strong** solution.

2. The gas phase. The words "gas" and "vapor" will be used more or less interchangeably, "vapor" being chosen when it is desired to emphasize a relation to a liquid phase. For a pure gas, the P-V-T (pressure-volume-temperature) relations can be obtained from the perfect gas law, or more accurately from an **equation of state,** or a table such as those in Chapter 7, representing actual measurements. If the solution contains more than one **volatile** component (absorbent as well as refrigerant volatile), or if evaporation is into a space which contains a foreign gas, **Dalton's Law** applies more or less accurately to the resulting gaseous mixture. Each component of a gaseous mixture exerts a partial pressure equal to the total pressure which it would exert if it alone were present, and the total pressure P is the sum of the partial pressures, P_1, P_2, etc., of the individual components.

Dalton's law is a corollary of the perfect gas law. For any perfect gas:

$$PV = nRT \quad \text{or} \quad P = \frac{nRT}{V}$$

Then for any component of a mixture of perfect gases:

$$P_1 = \frac{n_1RT}{V};\quad P_2 = \frac{n_2RT}{V};\ \text{etc}$$

and for the mixture:

$$P = (n_1 + n_2 + \cdots)\frac{RT}{V} = P_1 + P_2 + \cdots$$

Each n gives the number of mols of the gas indicated by the subscript, and R, the "gas constant," is the same for all perfect gases. $R = 10.71$ if P, the absolute pressure, is in psi, V is in cu ft, and T is in absolute F. Dalton's law may also be expressed in the form:

$$P_1 = N_1P;\quad P_2 = N_2P;\ \text{etc}$$

These relations follow from the above if one mol of mixture is taken, for which $P = RT/V$ and $n_1 = N_1$, $n_2 = N_2$, etc. This is the form in which Dalton's law is most commonly applied, but it is less general than the original statement, which permits the calculation of the partial pressures from actual P-V-T data on the individual components. If these are available and the concentrations are known, this form gives better accuracy than does the perfect gas law.

Dalton's law is the law of perfect gaseous mixtures. Mixtures of actual gases follow it fairly well, though not so well as the individual gases follow the perfect gas law, at least if the $P_1 = N_1P$ form is used. Accordingly, Dalton's law is useful, although not quantitatively accurate, up to fairly high pressures; how high depends on the particular mixture. It always breaks down as the partial pressure of any component and the temperature simultaneously approach the critical values for that component.

3. The **vapor pressure** of a liquid is the pressure at which vapor from the liquid is in equilibrium with the liquid. Applying Dalton's law, each component of a solution is said to have a definite **partial vapor pressure,** which is equal to its partial pressure in the equilibrium vapor. All of the partial vapor pressures, and therefore also the (total) vapor pressure and the equilibrium composition and concentration of the vapor, are physically determined by the temperature and composition of the solution; the quantitative relations depend on the particular solution. For a given composition of solution, the partial and total vapor pressures increase rapidly with temperature, following about the same rules as do the vapor pressures of pure liquids. For a given temperature, the partial vapor pressure of a solution is always less than the vapor pressure of the pure component, increasing with its concentration in solution. The equilibrium composi-

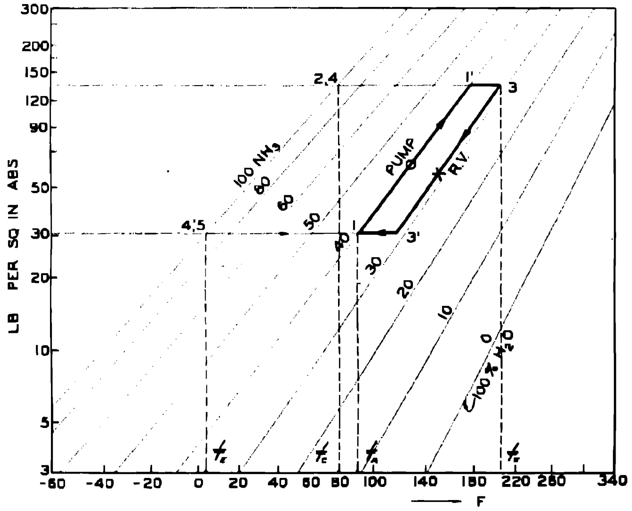


Fig. 2. Log p vs $1/T$ Chart for Ammonia Water

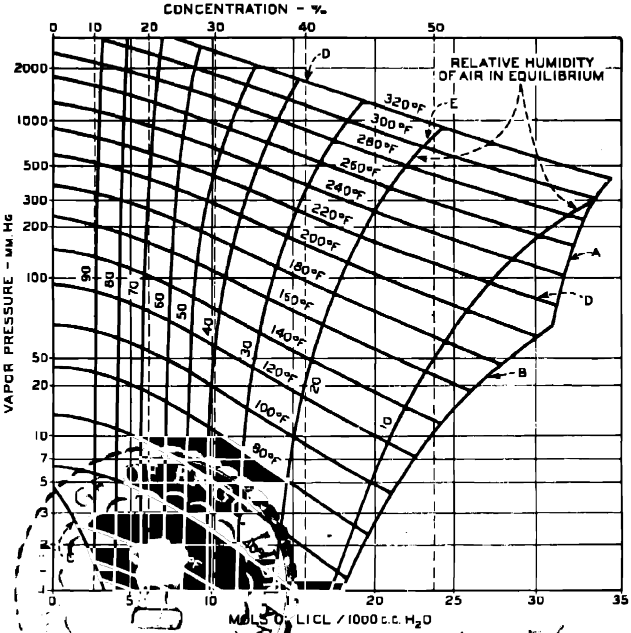


Fig. 3. Water-Lithium Chloride Equilibria

79823

601.56
A5R

tion of the vapor is in general different from that of the liquid, and usually is richer in the more volatile component.

Quantitatively, the relations between a solution and its vapor are complicated and must be determined experimentally in each case (for example, **Figs. 2, 3** and **4**), but there are two simple limiting laws which apply at the ends of the concentration range. **Henry's Law** applies to the minor component, or solute: The partial vapor pressure of a component whose mol fraction in the solution is sufficiently low, is proportional to the mol fraction:

$$P_1 = k_1 N_1$$

The value of the Henry's law constant, k_1, and the range of mol fractions over which it applies, depend on the particular solution and the temperature, and must be found by experiment. **Raoult's Law** applies to the major component, or solvent: The partial vapor pressure of a component whose mol fraction in solution is sufficiently near unity equals the product of the mol fraction and the vapor pressure of the pure solvent:

$$P_2 = N_2 P_2^0$$

Raoult's law may be considered to be a special case of Henry's law, in which the Henry's law constant is the vapor pressure of the pure component, P_2^0. For a given component, Henry's law applies at one end of the composition range, and Raoult's law at the other. There are no general rules for predicting the range of compositions over which either applies, nor what happens in between.

Perfect solutions, those for which Raoult's law applies to all components over the entire range of compositions, result when the molecular forces around the solvent and solute molecules resemble each other so closely that there are no special attractions or repulsions between solvent and solute molecules. For example, any pair of fairly closely related hydrocarbons may be depended on to follow Raoult's law quite closely, over the whole range of compositions and temperatures. If the components are much different chemically, their solutions are likely to deviate more or less from Raoult's law over most of the composition range. If the partial vapor pressure of a component exceeds the Raoult value at a given mol fraction, the deviation

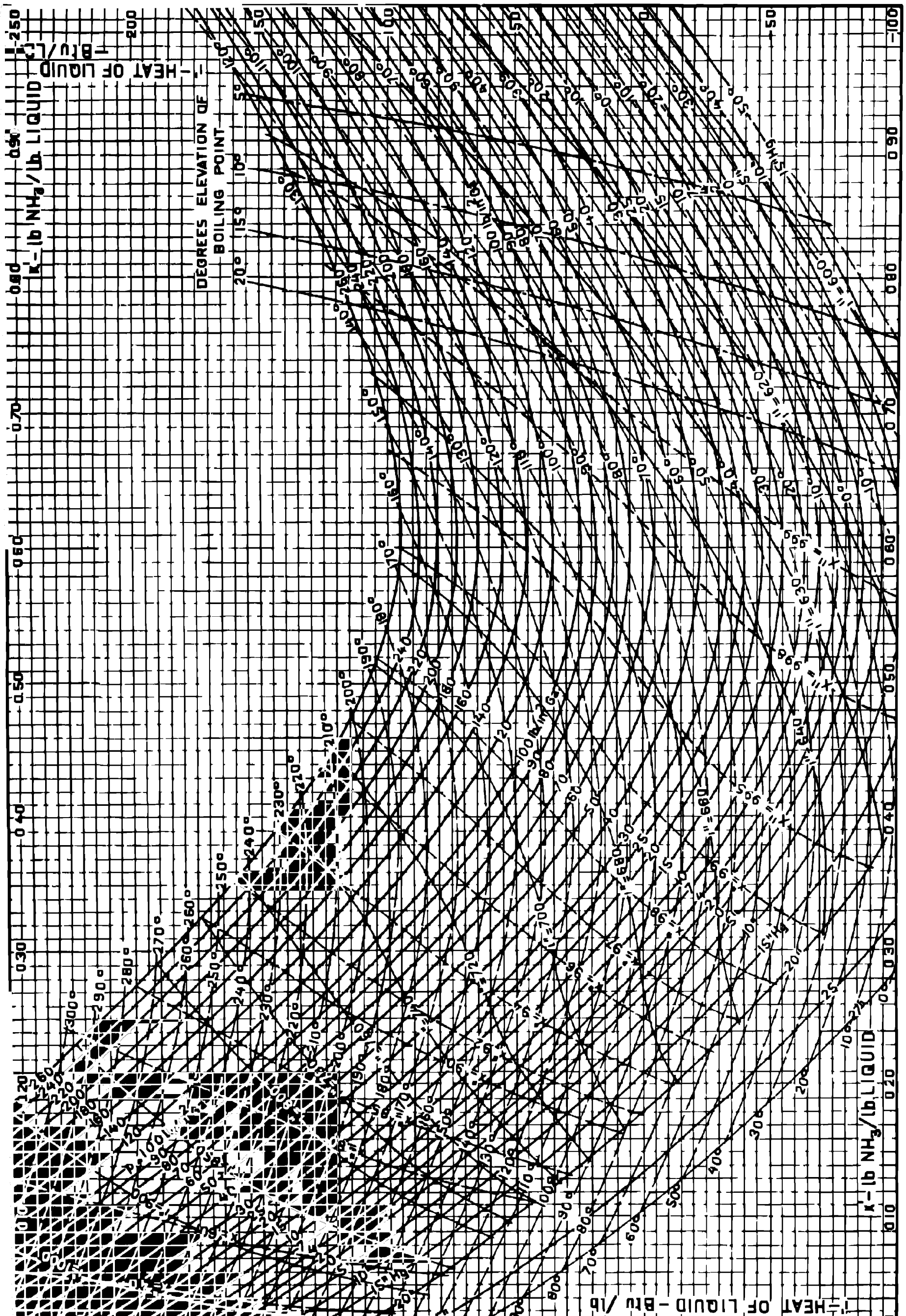


Fig 4. Enthalpy-Concentration Chart of Properties of Aqua Ammonia

is said to be positive, and the solubility is less than it would be if the solution were perfect. This is always the case when two liquids do not mix completely, and often when they do; it shows that the solvent and solute molecules repel each other.

If the partial vapor pressure of a component is less than the Raoult value, the deviation is negative, and the component is more soluble than it would be if the solution were perfect. This situation arises only in cases where there is special attraction between solute and solvent molecules. Usually, both components of a two-component system deviate from Raoult's law in the same direction, the deviations are in the same direction for all concentrations and temperatures, and the deviations become smaller percentagewise at higher temperatures; but there are occasional exceptions.

4. It is essential for the refrigerant to be highly **soluble in the absorbent,** at the temperature and partial pressure at which the absorbent is required to take up the refrigerant, in order to keep down the volume of absorbent required to be circulated, the size of the apparatus, and the heat losses. Large negative deviations from Raoult's law are therefore required; and the larger and more complicated the absorbent molecule, the larger the deviations which are required. On the other hand, the negative deviations must not be so great as to prevent recovery of a large proportion of the dissolved refrigerant by distillation, and must not result in the formation of an insoluble compound between the absorbent and refrigerant. It is not practical to set up definite rules, but an idea of the requirements may be obtained by comparison with the aqua-ammonia system, which is the one most commonly used in absorption refrigeration. In the relatively small-molecule H_2O-NH_3 system, the solubility of ammonia under absorber conditions is typically around four times the Raoult value, and is none too large at that. Very few other combinations of solvent and solute give as large negative deviations without forming insoluble compounds.

The requirements for a successful absorbent-refrigerant combination are extremely severe in numerous respects. The solubility requirements are hardest to meet. Another important consideration is that the absorbent must have a much higher boiling point than the refrigerant, so that practically pure liquid refrigerant can readily be obtained from the solution by fractional distillation. In this respect, the water-ammonia system is far from ideal, and few others are better. Also, irreversible chemical reactions of all kinds, such as decomposition, polymerization, and corrosion, are to be avoided. It is not surprising that extremely few acceptable refrigerant-absorbent combinations have been found.

5. The pure absorbent may be a solid; it is only the absorbent solutions which are required to be liquid. For example, a number of salts such as LiCl and LiBr may be used as absorbents with water as refrigerant, because they are extremely soluble in water and form **hygroscopic brines.** Such salts have the great advantages of being small-molecule, low-specific-heat, completely non-volatile absorbents, with very large negative deviations from Raoult's law within their operating ranges; but the operating ranges are limited by the appearance of solids. This limits the reduction in temperature attainable in the evaporator; and, in any case, a water evaporator must be operated above the freezing point of water. Hygroscopic brines are used to dehumidify air or other gases by direct contact, and also in absorption refrigeration. Properties of LiCl brines are shown in **Tables 1** and 2 and Fig. 3.

Table 1. Specific Heat of LiCl-Water Solution

Concentration, % of LiCl by weight	Specific heat, Btu/lb F
.5	.993
5.0	.960
10.0	.901
20.0	.808
30.0	.730
40.0	.660

6. Preparation of a usable **table** or **graph** from the usually quite limited amount of original data available on the liquid-vapor equilibrium of a given absorbent-refrigerant combination requires special procedures for correlation and interpolation. Several such procedures are familiar to chemical engineers. For fixed liquid composition and varying temperatures, the logarithms of the partial and total vapor pressures may be plotted against the reciprocal of the absolute temperature; or "Dühring's rule"[1] or Othmer[2] plots may be made. All of these give straight or nearly straight lines which relate the total or partial vapor pressure to the temperature. A skeleton log P-$1/T$ chart is shown for ammonia-water in Fig. 2. For fixed temperature and varying concentration, "Raoult's law" plots, for example, of P, P_1, P_2, $P_1/P_1^0\ N_1$ and $P_2/P_2^0\ N_2$ against N_1, can be recommended. The form of the final table or graph is largely a matter of taste; it can well be of some other type.

Table 2. Specific Gravity of LiCl-Water Solution
(Concentration is given in percentage of LiCl in mixture by weight)

% LiCl	Temperature, C								
	0	10	20	30	40	50	60	80	100
1	1.00604	1.00574	1.00411	1.00150	.99806	.99391	.9891	.9779	.9646
10	1.05966	1.05833	1.05594	1.05296	1.04954	1.04571	1.0414	1.0315	1.0199
20	1.11979	1.11789	1.11504	1.11181	1.08550	1.10500	1.1013	1.0927	1.0829
30	1.18464	1.18219	1.17914	1.17602	1.17269	1.16940	1.1659	1.1576	1.1507
40				1.24806	1.24427	1.24081			
45				1.28768	1.28361	1.28003			

Thermal data, i.e., total and latent heats, are needed in addition to equilibrium pressure-temperature-composition data, and should be correlated thermodynamically with the latter. Entropies are desirable also. Procedures will not be discussed here, but certain qualitative relations may be pointed out. As a general rule, the latent heats of vaporization of both refrigerant and absorbent are somewhat higher from solution than they are from the pure liquids, because of the large negative deviations from Raoult's law. The increase in latent heat is part of the price of better absorption. It then follows from thermodynamics that heat is evolved on mixing pure liquid refrigerant with pure liquid absorbent or with weak solutions of refrigerant in absorbent; and also that the partial and total vapor pressures of solutions increase more rapidly with temperature (i.e., the slope of log P-$1/T$ plots is greater) than do the vapor pressures of the pure components. A large heat of mixing of liquids (i.e., a large temperature rise) is an excellent indication of large negative deviations from Raoult's law and high solubility.

Charts giving the equilibrium pressure-temperature-concentration relations are shown in Fig. 2 for the ammonia-water system, and in Fig. 3 for the water-lithium chloride system. Fig. 4 shows the equilibrium pressure-temperature-concentration relations and the corresponding total heats of liquid and vapor, for the ammonia-water system. Fig. 4, prepared by Stickney,[3] is of a type suggested by Merkel and Bosnjakovic,[4] in which the coordinates are weight-concentration and enthalpy. A similar chart, for the two phases separately, was prepared by Berestneff.[5] Stickney's Fig. 4 superimposes the data for the liquid phase (index ′) and for the vapor phase (index ″). The use of Fig. 4 is illustrated by **Fig. 5,** which shows a complete set of readings for one point on Fig. 4. Fig. 4 gives no information concerning superheated vapor. The enthalpies (total heats) for liquid in equilibrium with vapor, also apply to subcooled liquid: that is, to liquid of the same composition and temperature, but under pressure in excess of the vapor pressure. The most recent aqua-ammonia tables are those of Scatchard et al in *Refrigerating Engineering* of May 1947.

II. Principles of Absorption Refrigeration

7. The **basic cycle** of liquid-absorbent refrigerating systems is indicated in **Fig. 1.** The four major steps are latent-heat processes involving the transfer of refrigerant between the liquid and vapor states, in the

evaporator, absorber, generator and condenser, respectively. Nearly pure liquid refrigerant boils in the evaporator, absorbing heat and producing refrigeration at a low temperature. The vapor passes to the absorber, where it is absorbed by weak solution, producing rich solution and evolving waste heat which is rejected to the cooling medium at ordinary temperature. The evaporator and absorber operate at the low-side pressure, which is determined by the vapor pressure of the refrigerant at evaporator temperature. Rich solution is pumped to the generator, where high-temperature heat is supplied, to boil out refrigerant vapor. The vapor then passes to the condenser, where it condenses and evolves waste heat which is rejected to the cooling medium at ordinary temperature. The generator and condenser operate at the high-side pressure, which is determined by the vapor pressure of the refrigerant at the condensing temperature. The weak solution produced in the generator is fed to the absorber through a valve, and the liquid refrigerant produced in the condenser is fed to the evaporator through another valve.

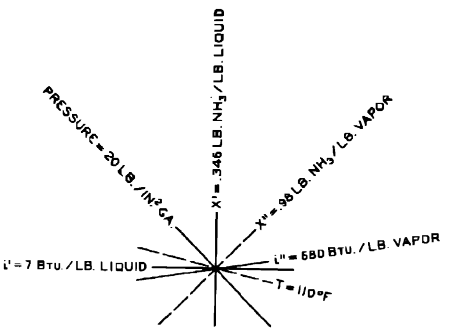

PRESSURE = 20 LB./IN.² GA.
X' = .346 LB. NH₃/LB. LIQUID
X" = .98 LB. NH₃/LB. VAPOR
i' = 7 BTU./LB. LIQUID
i" = 680 BTU./LB. VAPOR
T = 110°F

Fig. 5. Illustrating Use of Fig. 4

8. Consider the unrealizable ideal case for which the absorbent is non-volatile, the specific heats of both absorbent and refrigerant are zero, the four processes listed above are the only ones of thermodynamic significance, and these are carried out under conditions of perfect reversibility, with rich and weak solutions of the same concentration, namely, that for which the partial vapor pressure of refrigerant at absorber conditions equals the vapor pressure of pure refrigerant at evaporator temperature. The refrigerant vapor would obey the perfect gas law, and the latent heats would be independent of temperature at constant concentration, and for the ideal combination, as for any other, would presumably be slightly higher from the solution than from the pure refrigerant. Defining the cycle efficiency or performance factor *PF* as the ratio of the heat absorbed in the evaporator to the heat absorbed in the generator, the *PF* of the ideal cycle would be slightly less than one under all circumstances, decreasing slightly as the specified difference between evaporator and absorber increased. A simple thermodynamic expression for the ideal *PF* can readily be obtained. Identify the following quantities:

	Absolute temperature	*Heat absorbed or evolved*
Evaporator	T_1	Q_1
Absorber and condenser	T_2	Q_2
Generator	T_3	Q_3

Let w equal the theoretical work, for the ideal reversible cycle, which is derived from Q_3 supplied at T_3, and is used to remove Q_1 at T_1, the heat-sink being at T_2 for both. Then by the first law of thermodynamics:

$$Q_2 = Q_1 + Q_3$$

and by the second law:

$$w = Q_1 \frac{T_2 - T_1}{T_1} \text{ and also } w = Q_3 \frac{T_3 - T_2}{T_3}$$

Then

$$\frac{Q_1}{Q_3} = PF = \frac{T_3 - T_2}{T_2 - T_1} \cdot \frac{T_1}{T_3}$$

The latter expression is not to be regarded as giving theoretical values of *PF* which vary with the arbitrary choice of T_1, T_2, and T_3, because, as mentioned above, Q_1/Q_3 is fixed by other considerations and has a value of about 1; say 0.9. What it really means is that

$$(T_3 - T_2) = 0.9\,\frac{T_3}{T_1}\,(T_2 - T_1)$$

is approximately correct for the ideal relation between the three temperatures. The ideal $T_3 - T_2$ is thus of the same order as $T_2 - T_1$, usually somewhat smaller. Increasing $T_3 - T_2$ above the ideal value for a given $T_2 - T_1$ would not improve the theoretical *PF*, but it would provide an excess of available work over that ideally required, for use in overcoming irreversibility in the cycle. For example, it could be used to produce weaker solution and thus to provide driving head for the absorber process.

9. Actual cycles are never so good as the ideal cycle described above; whatever differences there are, are always of a sort which produce thermodynamic loss. Some of the processes waste high temperature heat, some waste refrigeration, and others waste available work in some other form. Both T_3 and Q_3 must be increased above the ideal values to compensate for these losses. There is considerable room for choice between increases in T_3 and Q_3; usually the optimum value of T_3 may be considered to be the one for which *PF* is a maximum.

The *PF* of actual absorption cycles has an upper limit of one, no matter how efficiently they are carried out nor how small the temperature differences between the evaporator and the condenser and absorber may be. On the other hand, the *PF* falls off only slowly as these temperature differences are increased, most of the additional available work required being obtainable through increases in T_3 instead of Q_3. These characteristics are very different from those of compression systems, for which the energy input per unit of refrigeration goes up in somewhat the same proportion as the temperature differences between the evaporator and the condenser, and the thermodynamic efficiency, instead of the performance factor, remains relatively constant.

10. Analysis of simple cycle. Design, operating characteristics and performance can be analyzed by setting up appropriate material and heat balances. For any chosen section of an absorption refrigerating system in a steady state, intake must equal outgo, for refrigerant, for absorbent and for the sum of the enthalpy of absorbent plus refrigerant and heat from external sources. Giving intake and outgo opposite signs, these conditions can be represented by the equations:

$$\Sigma W = 0; \quad \Sigma Wx = 0; \quad \Sigma(Wi + Q) = 0$$

where $W = G_1 + G_2$ = weight of absorbent plus refrigerant, lb
$Wx = G_1$ = weight of refrigerant, lb
i = enthalpy, Btu per lb
Q = heat added or rejected, Btu

Consider an aqua-ammonia system operating on the simple cycle shown in Fig. 1. The volatility of water under generator conditions is a major consideration in any aqua-ammonia system. The proportion of water in the vapor leaving the generator depends on the composition of the liquid and the temperature. In the simple cycle, no provision is made for removing water from the vapor before it enters the condenser, hence the water passes through the condenser to the evaporator; and being almost non-volatile under evaporator conditions, it concentrates as evaporation proceeds, and raises the boiling point of the remaining refrigerant. To keep the evaporator operating, the accumulated water must be purged to the absorber, either continuously or from time to time, as a liquid of high ammonia content.

Assume that vapor and weak solution leave the generator in equilibrium at chosen values of generator temperature and condenser pressure; that condensation is at equilibrium for vapor of the composition so determined; that rich solution leaves the absorber at a chosen temperature and at a composition such that its vapor pressure equals a chosen suction pressure; and that vapor and purge liquid leave the evaporator together, in equilibrium at the chosen suction pressure and a chosen outlet temperature. Assume the following conditions:

Generator temperature, F	220
Condenser pressure, psig	155
Suction pressure, psig	20
Temperature of suction gas and purge liquid leaving evaporator, F	20
Temperature of rich (or strong) liquid leaving absorber, F	90

To find the material and heat balances, prepare **Table 3**. From Fig. 4, find x and i for points 1, 2, and 3 in Fig. 1, and enter them in the table. For point 1, strong liquid leaving the absorber, the intersection of the 20 psi line with the 90 F line, gives for the liquid, $x = .405$ and $i = -16$. Points 2 and 3, gas and weak liquid leaving the generator, are found at the intersection of the 155 psi line and the 220 F line, which gives, for the gas, $x_2 = .920$ and $i_2 = 755$, and for the liquid, $x_3 = .316$ and $i_3 = 133$.

Table 3. Conditions in Cycle

Point	p	t	x	i	W	Wi
1	20	90	.405	−16	6.79	−109
2	155	220	.920	755	1.00	755
3	155	220	.316	133	5.79	770
4	155	92	.920	113	1.00	113
5	20	20	.920	414	1.00	414

Point 4 is liquid leaving the condenser and point 5 is the mixture of vapor and purge liquid leaving the evaporator. The material passing through the condenser and evaporator is that leaving the generator as gas, hence, on the customary basis of one pound of liquid leaving the condenser:

$$x_2 = x_4 = x_5 = .920$$

and

$$W_2 = W_4 = W_5 = 1$$

For point 4, at $x = .920$ for the liquid and a pressure of 155 psi, the chart gives $i_4 = 113$ and a condenser temperature of 92 F. For point 5, at 20 psi and 20 F, the chart gives, for the purge liquid, $x_5' = .753$ and $i_5' = -27$, and for the vapor, $x_5'' = .999$ (nearly) and $i_5'' = 622$. Neglecting the trace of water in the vapor, the weight of purge liquid is

$$W_2 = \frac{x_5'' - .92}{x_5'' - x_5'} = \frac{.079}{.246} = .321$$

Then i_5 for the mixture is:

$$i_5 = .321\ (-27) + (1 - .321)\ 622 = 414$$

For the generator, the material balances give:

$$W_1 - W_2 - W_3 = 0$$

and

$$W_1 x_1 - W_2 x\ - W_3 x_3 = 0$$

Substituting the previously determined values of W_2, x_1, x_2 and x_3, combining the two equations and solving:

$$W_1 = \frac{.920 - .316}{.405 - .316} = 6.79; \quad W_3 = 5.79$$

The Wi values are next filled in by multiplying together the W and i for each point, completing Table 3.

The following quantities may now be calculated. Per pound of liquid leaving the condenser:

Refrigerating effect $= i_5 - i_4 = W_5 i_5 - W_4 i_4 = 301$ Btu
Heat input to generator by steam $= W_2 i_2 + W_3 i_3 - W_1 i_1 = 1{,}634$ Btu
Heat removed from condenser $= W_2 i_2 - W_4 i_4 = 642$ Btu
Heat removed from absorber $= W_3 i_3 + W_5 i_5 - W_1 i_1 = 1{,}293$ Btu

To get the refrigerating effect in tons, per pound of liquid leaving the condenser **per minute**, divide 301 by 200 which equals 1.5.

The ratio of refrigerating effect to heat input is the performance factor, PF, of the cycle:

$$PF = \frac{301}{1634} = .184$$

This is decidedly low in comparison with the ideal PF of around .9 obtained in paragraphs 8 and 9. The largest losses are those due to heat carried away from the generator by the weak liquid, and to the water volatilized in the generator and carried into the evaporator. Additional equipment such as described later reduces the losses.

III. Industrial Absorption System

1. Conventional machine. The simple ammonia-water cycle as described previously is very uneconomical and can be

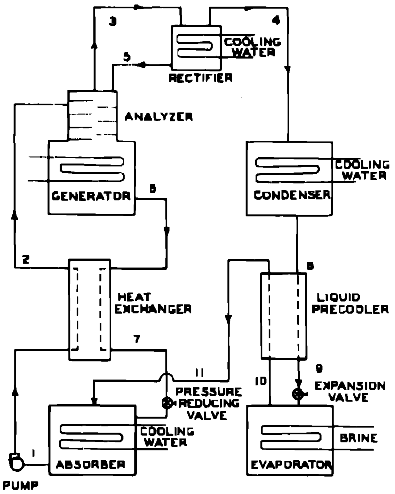


Fig. 6. Simple System with Auxiliary Apparatus

greatly improved by auxiliary apparatus. Some of the auxiliary equipment described in the following paragraphs may be omitted in actual application, depending on the combination that will give the best economic balance between first cost and operating cost.

Two parts for reducing the amount of water in the condenser and evaporator are in general use: the analyzer and the rectifier. The **analyzer** consists of a chamber through which the vapors leaving the generator pass in counterflow contact with the strong liquid coming to the generator. Either bubble cups or packing may be used in order to bring the vapor into better contact with the liquid. As the vapor passes upward through the analyzer it is cooled and enriched by ammonia, and the liquid is heated. Thus the vapor going to the condenser is lower in temperature and richer in ammonia, and the heat input to the generator is decreased. The ammonia concentration of the vapors can be increased still further by refluxing above the point where the strong liquid is introduced, but this will increase the heat input to the generator.[1]

The **rectifier** is simply the inlet part of the condenser arranged in such a manner that its condensate drains back to the generator or analyzer. The rectifier may be cooled by water or by strong liquid.

It is universal practice to use a **counterflow heat exchanger** for the strong liquid flowing from the absorber to the generator and for the weak liquid flowing from the generator to the absorber. The strong liquid leaves the absorber far below its boiling point and the weak liquid must be cooled before it enters the absorber; therefore the liquid heat exchanger permits a substantial reduction in input and saves appreciably on cooling water.

Liquid ammonia passing from the condenser to the evaporator is usually cooled by bringing it into **heat exchange** with the **suction gas** leaving the evaporator. Since a cooler liquid allows greater refrigerating effect per unit of weight, this heat exchanger gives additional refrigeration at no cost except a slightly higher temperature cooling water from the absorber.

The **weak liquid precooler** is a water-cooled section for cooling the weak liquid

Table 4. Conditions in Cycle

Point	p	t	x	i	W	Wi
1	20	90	.405	−16	7.64	−122
2	155	187	.405	94	7.64	720
3	155	187	.963	716	1.08	773
4	155	130	.993	667	1.00	667
5	155	130	.600	53	.08	4
6	155	220	.316	133	6.64	882
7	155	108	.316	6	6.64	40
8	155	86	.993	136	1.00	136
9	155	20	.993	61	1.00	61
10	20	10	.993	580	1.00	580
11	20	76	.993	665	1.00	655

before it enters the pressure-reducing valve. It has nearly the same effect as the abovementioned liquid gas precooler and may be installed in addition to the latter.

Stickney[6] and others[7] have worked out the effect on the heat balance of the addition of each of the above auxiliary devices to the simple cycle, and also their effects in combination. **Fig. 6** gives the hook-up of a machine employing some of them and **Table 4** gives the heat balance figures for this cycle, based on the following conditions and arrived at by the means explained previously.

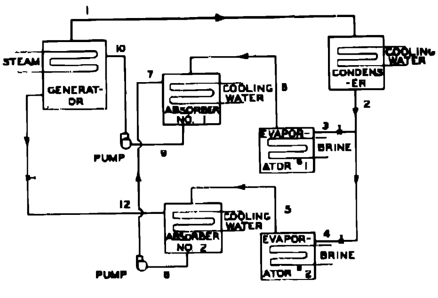


Fig. 7a. Two-Stage System

Suction pressure, psig	20
Suction temperature, F	10
Temperature of strong liquid from absorber, F	90
Generator temperature, F	220
Condenser pressure, psig	155
Temperature of vapors leaving rectifier, F	130

The liquid heat exchanger is of such a size that the strong liquid leaving it is saturated. The rectifier and liquid gas precooler are of such a size that the liquid ammonia is cooled to 20 F and the suction gas is heated to 10 F below liquid ammonia temperature leaving the condenser.

From the table the following quantities can be obtained:

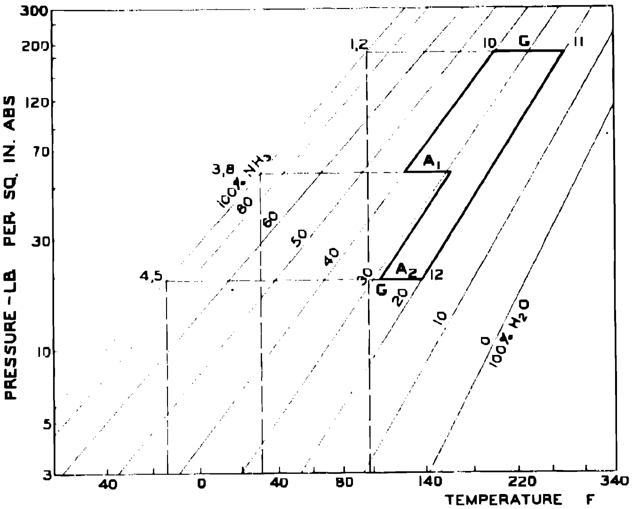


Fig. 7b. Cycle for Two-Stage System

Refrigerating effect,
$(W_9 i_9 - W_{10} i_{10})$ = 519 Btu
Heat input to the generator
$(W_2 i_2 - W_3 i_3 + W_8 i_8 - W_6 i_6)$ = 931

Total heat to the system = 1,450

Heat rejected in the absorber
$(W_7 i_7 + W_{11} i_{11} - W_1 i_1)$ = 817
Heat rejected in the condenser
$(W_4 i_4 - W_8 i_8)$ = 531
Heat rejected in rectifier
$(W_3 i_3 - W_4 i_4 - W_5 i_5)$ = 102

Total heat rejected = 1,450

All the above figures are per pound of the liquid from the condenser. The cycle efficiency of this process is then 519/931 = 55.6%. Thus it is seen that the auxiliary apparatus used in this machine has increased the efficiency almost three times as compared with the simple ammonia-water cycle described in Part II. At the same time the temperature in the evaporator is reduced by 10 F for the same suction pressure, which is also a desirable feature.

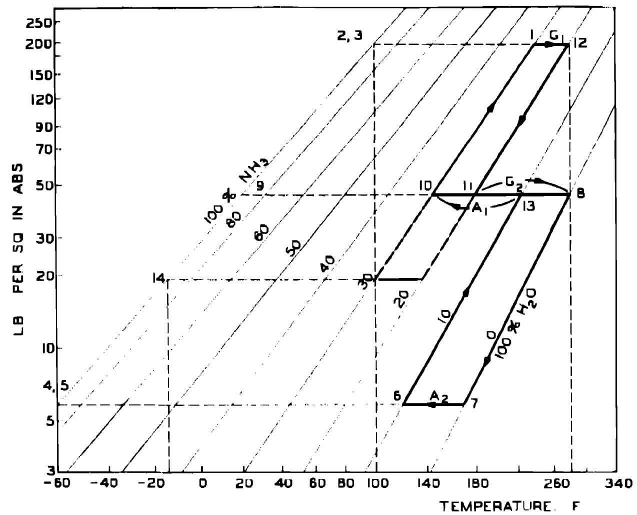
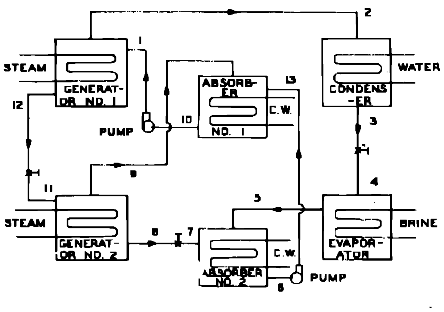


Fig. 8a (left). Cycle for System of 8b

Fig. 8b (below). Another Type of Two-Stage Ammonia-Water Absorption System



(More detailed description of the ammonia-water absorption machine can be found in reference 8.)

2. Multi-stage absorption machines. An absorption system may consist of two or more stages at different pressure levels. One arrangement, omitting for simplicity the auxiliary apparatus, is seen in **Fig. 7a.** There is one common generator and one common condenser for both stages, but special evaporators and absorbers for each stage. The flow of vapor and liquid are clear from the chart, whereby the cycle of Fig. 7b can be easily followed by using the reference numbers in Fig. 7a identifying the characteristic points. Cooling water is used in series through the condenser, low-stage absorber 2, and high-stage absorber 1. Accordingly, all points on Fig. 7b are located from left to right. The conditions at different points of the cycle can be found and the balance equations can be set up in the same manner as before.

Another example of a two-stage machine is shown in **Fig. 8b.** Here are two generators and two absorbers, one for each stage, but only one condenser and one evaporator. The advantage of this arrangement is that it permits a lower evaporator temperature at the same temperatures of the heating medium and cooling water, but at the same time it almost doubles the heat requirement. (Some other arrangements of the multi-stage machine can be found in reference 9.)

3. Resorption system. In the processes considered above, the evaporator and condenser are at constant temperature, which corresponds to the Carnot cycle. Lorenz[10] recognized and appreciated the fact that the cycle can be improved thermodynamically by using changing temperatures in the evaporator and condenser of a system similar to the changing temperature conditions taking place in the generator and absorber of the system. By arranging a counterflow between the solution leaving and entering the apparatus, the heat may be regained and the processes made more efficient.[11]

This cycle is represented in **Fig. 9a** and **b.** It differs from the absorption cycle, in

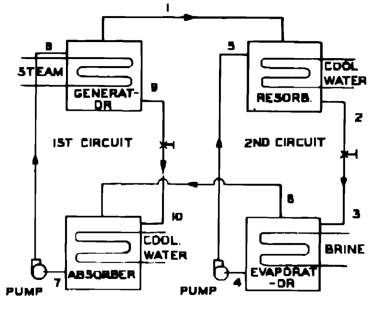


Fig. 9a. Simple Resorption System

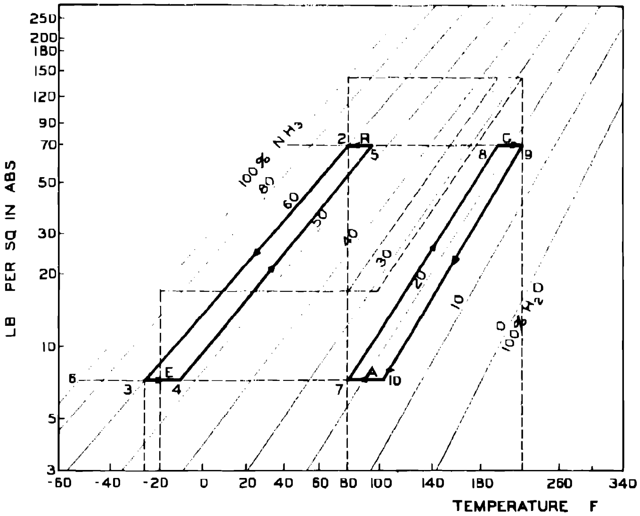


Fig. 9b. Cycle for System of Fig. 9a

that after being boiled off in the generator, the refrigerant vapor is not condensed to a pure liquid as at point 1, but is absorbed by a special weak solution passing through the apparatus called a resorber, points 5, 2.

As the concentration changes, the temperature changes as well. Cooling water going through the resorber in a closed circuit removes the heat of condensation and absorption. From the resorber the solution goes to the evaporator through the pressure-reducing valve. There it flows in counterflow with the brine, picks up heat which boils out the refrigerant. Again, as the concentration changes the temperature changes.

The resorber and evaporator are thus fed by the special solution circuit, requiring an extra pump. Against this there are advantages of the cycle in that it can produce more cooling effect at a lower level than the usual absorption system (see the cycle shown on the dotted line), with the same temperature of condensing water without a higher pressure. It is necessary thereby to go to a lower pressure on the low side. By changing the concentration of the resorber circuit, it is even possible to reduce the high-side pressure and still reach the desired temperature in the evaporator. From the above, it follows that the absorption system may be considered a particular case of the resorption system with 100% concentration of the resorber circuit.

The resorption and absorption systems can be combined in different ways. **Fig. 10** shows one combination, which results in almost doubling the refrigerating effect of the system, as the same heat is required to run the single effect (shown by the dotted line on Fig. 10a) and the double effect of the combined system. The pressure conditions must be selected properly. Note that more equipment is required, raising the first cost of the system.

Another combination for getting a low refrigerating temperature with a high condensing temperature, and a comparatively low temperature of the heating medium, was suggested by Maiuri.[12]

IV. Absorption for Air Conditioning

The absorption system may be applied to air conditioning by using it as a closed or an open system. The closed system cools and dehumidifies air by means of a cold contact surface in the same way as a compression system. The open absorption system and the open adsorption systems, which are closely related, are both known as dehydration systems. They operate by bringing the air into direct contact with

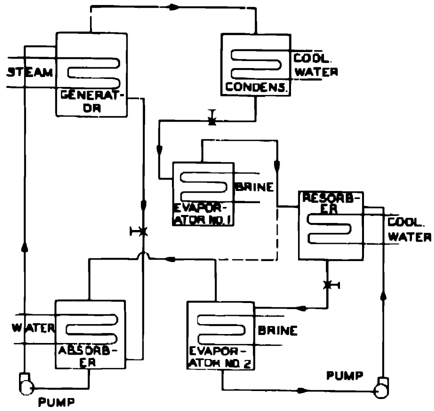


Fig. 10a. Combination Absorption-Resorption System

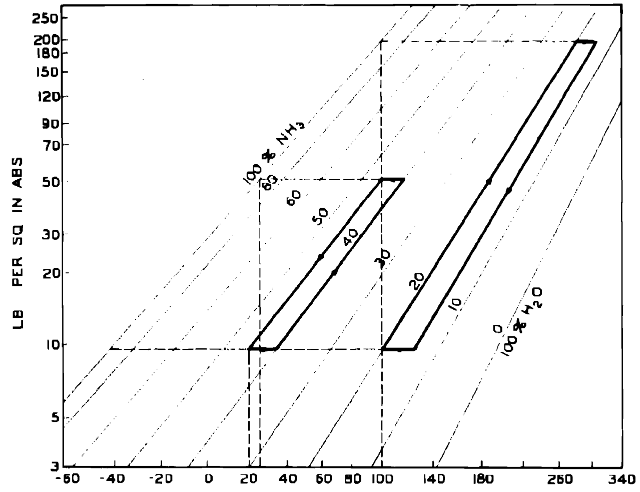


Fig. 10b. Cycle for System of Fig. 10a

the absorbent or adsorbent which has a high affinity for moisture. This dehydrates the air but also raises its temperature, therefore this sensible heat must be removed by means of a dry-air surface cooler. The air may be further cooled by evaporating water into it, provided the dehydration in the first step is extensive enough. According to present information the following units are available.

1. **Servel unit.** This unit is a closed absorption system using water as the refrigerant and lithium bromide solution as the absorbent. It is an all-year air conditioner having cooling units of 3 and 5 ton capacity with gas inputs of 69,000 and 112,000 Btu per hr, and having heating sections of 120,000 and 180,000 Btu per hr gas input ratings. Operation of the unit is best explained by referring to **Fig. 11.**

A 25 ton chilled water type unit, using the same cooling cycle, is also available.

No solution pump is used for circulating the solution. Instead, refrigerant vapor is produced in the generator in several vertical tubes, surrounded by a steam jacket, so that vapor lifting of the weak (in refrigerant) solution is obtained. From the separator at the top of the generator the refrigerant vapor passes to a water-cooled condenser from which it flows through an orifice to the evaporator. This orifice serves to maintain the pressure difference between condenser and absorber, pressure in the former being approximately 1 psia and in the latter approximately 0.15 psia. The evaporator is of the direct-expansion type in which the refrigerant is evaporated inside extended surface coils, with air passing over the outside surface. From the separator weak solution flows by gravity (and by pressure difference) through the liquid heat exchanger to the absorber. Here refrigerant vapor is absorbed and the strong solution returned by gravity through the liquid heat exchanger to the generator leveling chamber.

The water-cooled purge pump serves to collect non-condensible gases from the absorber and deliver them to a gas storage chamber.

A special boiler supplies steam under atmospheric pressure to the cooling unit generator when refrigeration is desired and to a heating coil when heating is desired. More detailed descriptions of this system may be found in references 13 and 14.

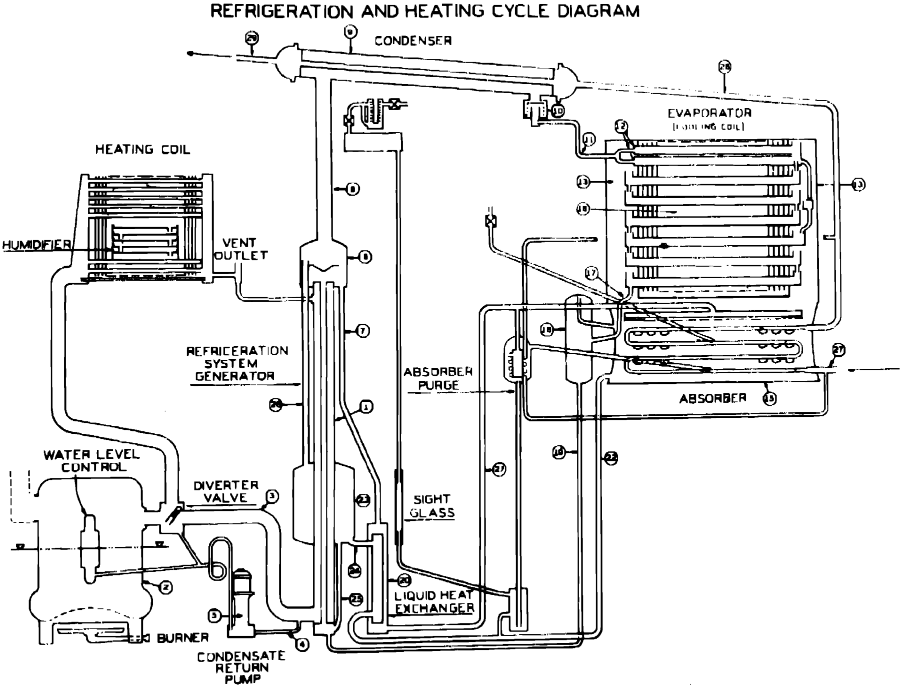


Fig. 11. Diagram of Servel All-Year Air Conditioning Unit

The cycle efficiency of the absorption cooling unit is approximately 60%.

2. **The Williams Air-O-Matic unit** is a closed absorption system using methylene chloride as refrigerant and dimethyl ether of tetraethylene glycol as absorbent.[15,17] With ratings of 7, 15 and (perhaps) 32 tons it is suitable for large residences and small commercial applications. It is a chilled water[15] type unit and requires a supply of low pressure steam. The boiler is not integral with the refrigeration unit.

The operation of this unit may be seen by referring to **Fig. 12.** Low-pressure steam enters the steam space of the heater where it comes in thermal contact with coils containing solution strong in refrigerant. Heated strong solution is discharged from the coils into an inner vapor release chamber where refrigerant vapor separates from the solution at a pressure of 1 to 3 psig. The vapor passes from the top of the release chamber down to the condenser, from which refrigerant liquid is discharged through a drain pipe and passed through the absorbent liquid-return heat exchanger, then through the refrigerant float valve into the water chiller. Here it is vaporized at low pressure (21 to 24 in. mercury vacuum) by absorbing heat from water circulating to the air conditioner.

Weak solution remaining after refrigerant vapor is expelled in the inner vapor release chamber flows from the bottom of the chamber through the liquid heat exchanger, and thence to the top of the absorber where it is sprayed over cooling coils. Refrigerant vapor from the water chiller passes up through a baffle plate to the top of the absorber and is absorbed by the solution passing down over the cooling coils. Strong solution from the bottom of the absorber is pumped by the solution pump through the liquid heat exchanger and on to the heater coils in the steam space.

Liquid refrigerant entering the water chiller contains a small amount of absorb-

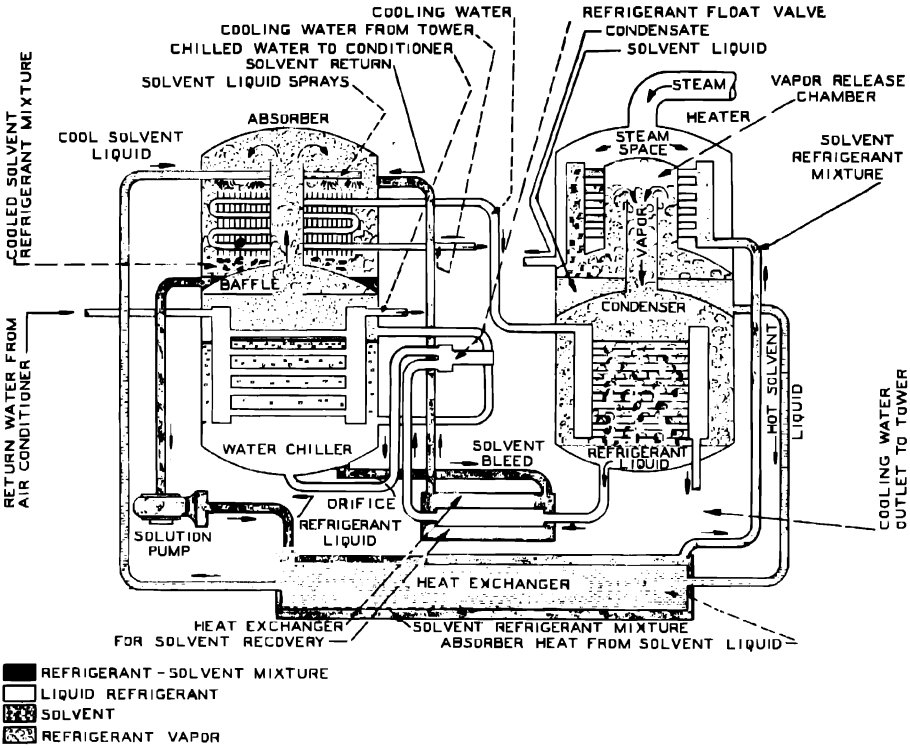


Fig. 12. Williams Air-O-Matic Absorption Unit

ent that must be removed from the chiller. This is done by continuously bleeding off a little liquid through an orifice at the bottom of the chiller and passing it in heat exchange with liquid refrigerant entering the chiller. Part of the refrigerant in the bleed-off mixture is evaporated in the heat exchanger and raises the remaining solution to the top of the absorber.

The cycle efficiency of the latest make, according to the manufacturer's catalog, is approximately 50 %.

3. The **Kathabar unit** for dehumidifying air, is an open absorption system employing Kathene (lithium chloride brine) as the absorbent. According to the flow chart of **Fig. 13,** circulated air is passed through a spray of Kathene. Moisture is absorbed from the air by the brine which then flows to a regenerator where the solution is regenerated by heating it and passing outside air over it to carry off the excess water vapor.

If insufficient moisture is present in the circulating air the system may be reversed and moisture given up to the air by the brine. Whether moisture is added to or removed from the circulated air depends on the setting of an automatic humidistat control.

A series of four standard self-contained units, including contactor and regenerator section, and with capacities of 750, 1,500, 3,500, and 5,000 cfm, is available. For build-up systems contactor cells are made in two sizes having air capacities of 3,000 and 4,000 cfm. For large installations these cells are combined in batteries and regenerating systems are sized as required to handle these batteries.

4. The **Bryant unit** is an open adsorption unit using silica gel as the adsorbent. This

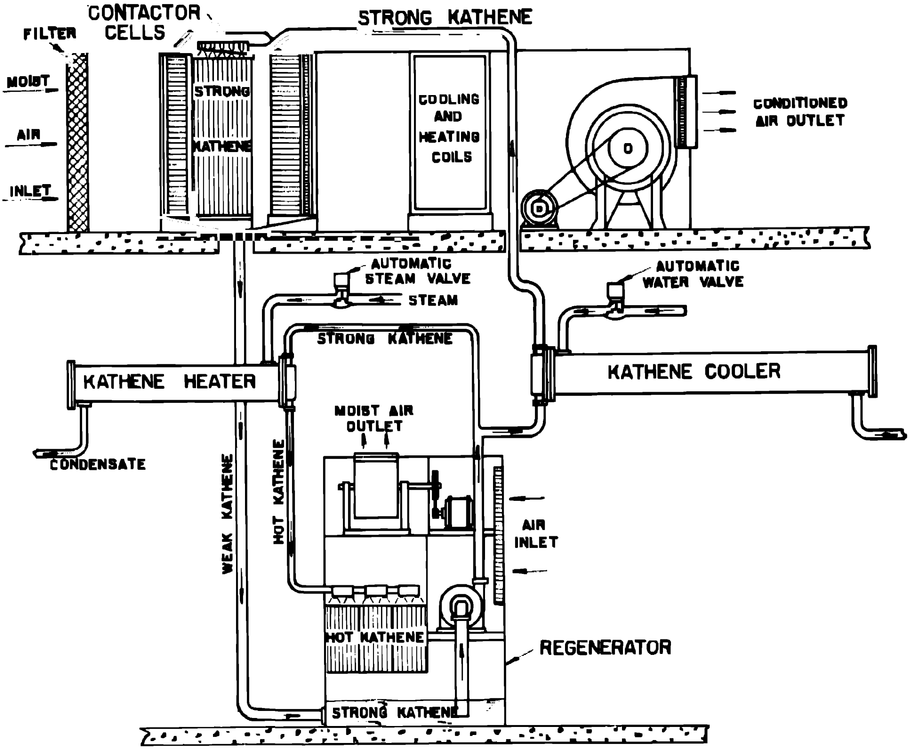


Fig. 13. Air Dehumidification under Summer Conditions with Kathene Solutions

unit, available in four sizes with nominal air capacity ratings of 500, 800, 1,300 and 2,900 cfm, may be used for commercial and residential air conditioning as well as for industrial processes. For comfort cooling purposes, companion dry air coolers and resaturators are obtainable from the manufacturer.

The cycle is illustrated in **Fig. 14**. A hollow revolving drum constructed of two concentric cylinders of fine mesh screen contains the silica gel in granular form between its walls. Interior of the drum, as well as surrounding space, is divided by means of wiper blades into two comparatively airtight sections. The rest of the cycle is apparent from the illustration. The drum is rotated at a rate of approximately one revolution in seven minutes. In the resaturation cooler air is forced through a wetted rotating drum made of large mesh screen.

In addition to the above unit, Bryant makes two sizes of stationary bed-type dehumidifiers which have air capacities of 2,700 and 5,000 cfm.

5. The **Lectrodryer unit** is an open adsorption unit using activated alumina as adsorbent. The manufacturer specializes primarily in air and gas drying equipment for all sorts of industrial processes and produces standard units in sizes ranging from 350 to 15,000 cfm air capacity. **Fig. 15** shows the arrangement of one type of Lectrodryer unit. Air to be dehumidified and heated reactivation air pass alternately through beds of activated alumina. If cooling is desired in addition to dehumidification, water coils are supplied, either imbedded in the alumina beds, or installed in outlet air ducts.

6. The **Carrier unit** is a closed absorption system using water as the refrigerant and lithium bromide solution as the ab-

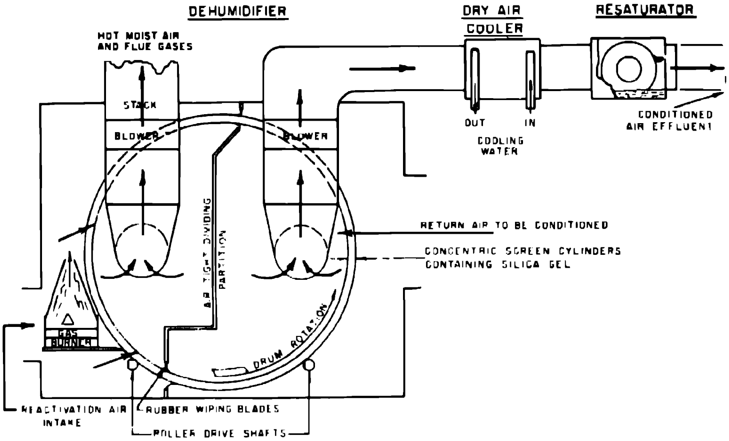


Fig. 14. Flow Diagram of Bryant Dehumidifier with Dry Air Cooler and Resaturator

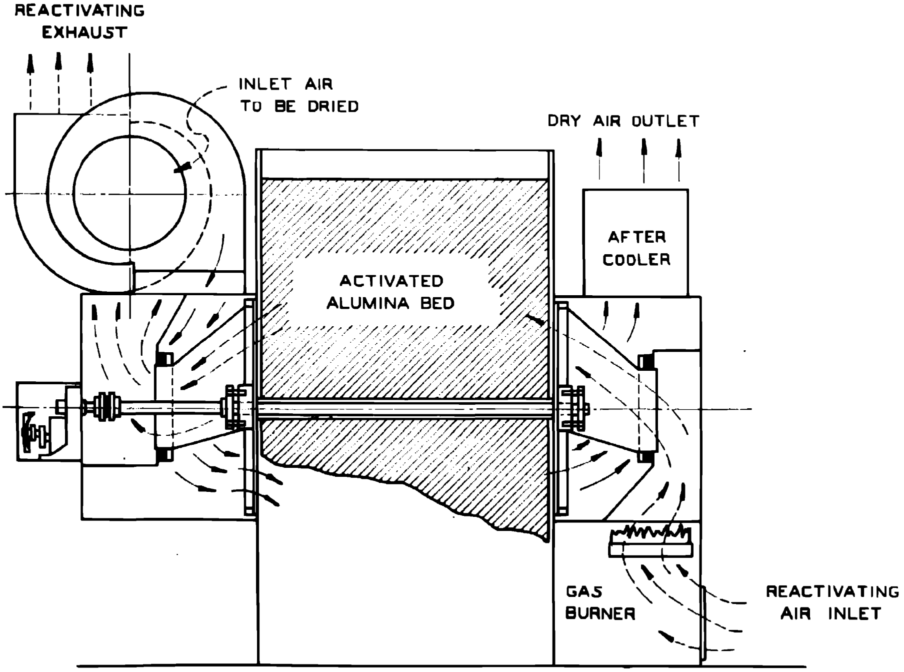


Fig. 15. Schematic Diagram of Pittsburgh Lectrodryer Using Activated Alumina as Drying Agent

sorbent. The machine is designed for chilled-water application, and is of the type that circulates the refrigerant water as the cooling medium through the air treating coils. The unit is available in capacities of 115, 150, 200, 270 and 350 tons.

The arrangement and operating cycle of the unit are shown in flow diagram of **Fig. 16.** (For more detailed information see reference 18.)

Water to be chilled enters through header (1) and is sprayed into flash evaporator (2). Shell (3) is maintained at a low absolute pressure nearly corresponding to the temperature to which the water must be cooled. Part of the water flashes, cooling the remainder. It then drains to the suction of the pump (4) which delivers it back to the load. The corresponding pressure in shell (3) is a function of the concentration and temperature of the solution sprayed over the absorber coil (5). The flashed water vapor moves down and is absorbed by the solution. The latter is cooled continuously by the flow of condensing water passing through coil (5) to remove the resulting heat of condensation and dilution.

The rich solution is drained from shell (3) into the solution pump (6). This pump delivers a portion of the solution through the heat exchanger (7) to the generator coil (8), located in the upper shell (9). Steam admitted to the tubes boils off the water vapor previously picked up by the solution in the absorber.

Water vapor boiled out of the solution is liquefied in the condenser coil (10) by the condensing water passing inside the coil. The vapor condensate is returned to the evaporator (2) through loop (11) which acts as a seal against the difference in pressure between the evaporator and the condenser.

The weak solution from the generator passes through the heat exchanger (7), and enters into a liquid eductor (12). Here it is entrained by the other portion of the solution from pump (6) and is delivered back to the absorber spray which completes the cycle. The split arrangement reduces the flow of solution to and from the generator, and therefore the amount of heat to be exchanged in the heat exchanger, thus securing higher efficiency of the cycle.

Both shells are maintained at a high vacuum, one corresponding to the temperature of chilled water, the other to the condensing temperature. To remove air and other non-condensables, a two-stage purge (13) is provided consisting of steam and water jets with an interstage purge condenser. Two purge lines, one from the absorber and another from the condenser, connect the purge with the shells.

The solution pump is a small centrifugal pump driven by a medium-speed motor of 5 to 10 hp, depending on the capacity of the unit.

The efficiency of the unit, according to published data[18] is approximately 70% at 100% capacity.

V. Costs of Systems

From past experience it has been known that at high evaporating temperatures, possibly 0 F and up, the first cost of an absorption system is higher than that of a

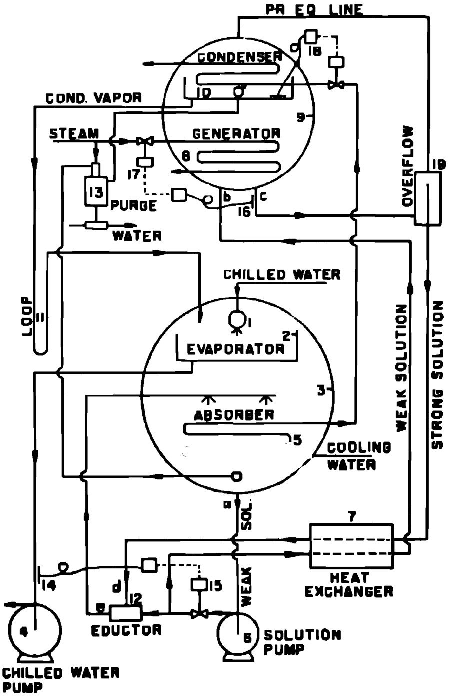


Fig. 16. Flow Diagram for Carrier Absorption Unit

compression system. At lower temperatures the absorption system is comparable or even cheaper, especially when compared to reciprocating machines. Yet results obtained in recent development indicate that a well-designed absorption machine can be built inexpensively for any temperature.

At a high temperature, say 45 F, the overall efficiency of the absorption machine, including boiler losses, can be made as high as 60% for one-stage evaporation. This value is to be compared with the efficiency of transforming heat into electrical energy, a maximum at 30% with a practical value of between 20 and 25% multiplied by the coefficient of performance of the compression cycle, of about 4.5. The product is .25 × 4.5 = 1.125.

Thermodynamically the compression system seems to be much better, but this is only a part of the story of operating cost, because to the cost of fuel some fixed charges must be added, which are considerably greater for the power plant than for a steam boiler or a direct-fired generator of the absorption system. If this is taken into consideration, operating cost of the absorption system will compare favorably with the compression system even at high temperatures. The cost of the absorption machine, both first and operating, is less influenced by the evaporating temperature than that of the compression machine, so that at low temperatures the absorption machine will compare still more favorably.

Other important points are: (1) The absorption machine has no prime moving parts; (2) no oil is needed at any point in this machine; (3) the machine is inherently easy to control at partial load, maintaining its full load efficiency; and (4) the machine is very adaptable to any set of working conditions, and can be fed with a low-pressure steam often available as waste.

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4. PSYCHROMETRY

I. Properties of Air

THE **properties of air** discussed here are presented in tabular form in Chapter 10, which should be referred to when tabular values are needed.

1.1—Dry Air Properties

Volume. Except for small, local variations in the fraction of carbon dioxide, dry atmospheric air has practically a constant composition from sea level to about 50,000-ft altitude as follows:

	Percent by volume	*Composition by weight*
Nitrogen	78.03	75.47
Oxygen	20.99	23.19
Monatomic gases, principally argon	0.94	1.30
Carbon dioxide	0.03	0.04
Hydrogen	0.01	0.00

Its equivalent molecular weight, derived by the laws of gaseous mixtures, from the above composition, is 28.966. Dividing this figure into the universal gas constant, 1545.3 ft-lb per mol deg F, results in a corresponding gas constant, R_a, of 53.349 ft-lb per lb deg F. The ideal gas equation of state resulting is:

$$v_a = \frac{R_a T}{p} = \frac{53.349T}{p} \qquad (1)$$

$$v_a = \frac{0.75430T}{p} \qquad (1.1)$$

Where v_a = specific volume of dry air (cu ft per lb)

T = absolute temperature, $t + 459.7$ (deg F abs)

p = absolute pressure, psfa in Equation (1) and in. Hg in Equation (1.1)

The volumes of dry air at standard atmospheric pressure given in the psychrometric tables of Chapter 10 are derived from a modified form of the Beattie-Bridgeman equation and are more exact than those which result from Equations (1) and (1.1). The deviations are small, being of the order of 0.25% at −100 F and less than 0.1% from 32 F to 200 F.

Enthalpy. Previously, the specific heat of dry air was taken as a constant, 0.24, so that enthalpy above 0 F could be expressed as $h_a = 0.24\ t$. Extension of the data to higher temperatures has made it imperative to recognize the **variation** of the **specific heat** of **air** with **temperature.** It seems to be generally accepted that values of the zero pressure specific heat of dry air derived from the zero pressure specific heats of its constituent gases, as calculated from spectroscopic data, are more accurate than values derived from direct thermal measurements.

The constant pressure specific heat at other pressures may be derived from the zero pressure data and an equation of state like the Beattie-Bridgeman equation using exact thermodynamic relations. This work of others was used in constructing **Fig. 1,** as follows: The zero pressure curve is a mean plot of the values given in Cornell University Experiment Station Bulletin No. 30,[1] and the Sage and Lacey tables.[2] A cross plot of the graphical data in Bulletin No. 30 indicated that c_p was essentially a linear function of pressure in the range of temperatures from 0 F to 200 F. This information, together with the correlated zero pressure data, served to locate accurately the curves of c_p at standard atmospheric pressure and the other higher pressures. Values of c_p at 29.921 in. Hg abs, calculated from the Goff and Gratch tables,[3] are indicated, providing a good check of the standard atmosphere line.

It has been noted above that the variation of specific heat with pressure is related to the fact that the volume of air does not exactly follow the simple equation of state (Equations (1) and (1.1)). In spite of this apparent inconsistency, the use of the specific heat data in Fig. 1 will improve the accuracy of computations at low temperatures and high pressures, even though the volumes used in the reduction of air flow data are derived from the simple equation. For exceedingly precise work, the Beattie-Bridgeman equation[1] may be used in place of Equation (1) for computing the volume of dry air.

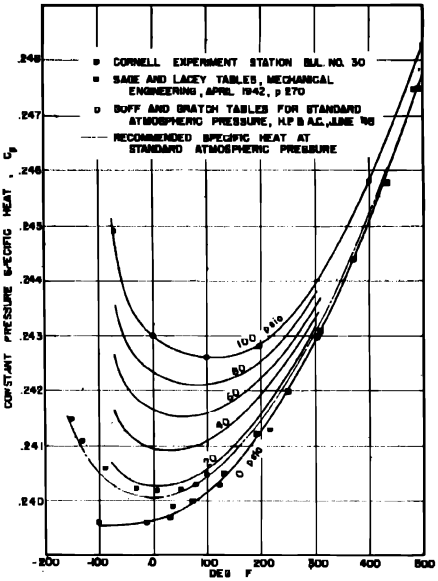


Fig. 1. Constant Pressure Specific Heat of Dry Air

1.2—Water Properties

Vapor pressure. The most accurate and consistent vapor pressure data available are apparently those developed by Goff and Gratch.[4] Below the triple point temperature and pressure (32.0176 F and 0.18167 in. Hg, respectively), a metastable liquid state exists. In other words, pure liquid water resists freezing if no ice crystals are present to initiate the process. The vapor pressure of liquid water below the triple point is higher than that of ice at the same temperature. This, it will be seen, results in two possible values for certain of the psychrometric properties of moist air. In the tables, emphasis is given the properties corresponding to the vapor pressure of the stable ice phase.

Enthalpy, h_g. The enthalpy of saturated water vapor, h_g, is used in preparing moist air properties. An accurate value becomes of increasing importance at higher temperatures where more water vapor may be contained by the air. In preparing the psychrometric tables, values for h_g, in Btu per lb of steam, as given in the tables of water vapor properties,[4] were used, but the following empirical equations in terms of Fahrenheit temperature, t, will generally provide satisfactory accuracy:

For the range $t = -100$ F to 70 F
$$h_g = 1061.7 + 0.439\,t$$
For the range $t = 70$ F to 200 F
$$h_g = 1063.9 + 0.410\,t$$

Similar equations for the enthalpy of liquid water and ice are:

$$h_f \text{ (for water)} = (t - 32)$$
In the range 32 F to 212 F

$$h_f \text{ (for ice)} = -159 + 0.48\,t$$
Below 32 F

Specific gravity referred to air, G. By definition, the specific gravity of water vapor referred to air is the ratio of the densities of water vapor and dry air at the same temperature and pressure. Alternatively, it may be expressed as the ratio of specific volumes which, of course, are simply the reciprocals of the densities. Thus

$$G = \left(\frac{v_a}{v_w}\right)_{p_w, T} = \frac{53.349}{R_w} \qquad (2)$$

Where G = specific gravity of water vapor referred to air
v_a = specific volume of dry air from Equation (1) at pressure p_w and temperature T
v_w = specific volume of water vapor at its own partial pressure, p_w, and the absolute dry-bulb temperature T

$$R_w = \frac{p_w v_w}{T}$$

Since water vapor does not follow the ideal gas laws exactly, R_w is a variable depending on pressure and temperature, resulting in a corresponding variation in G as shown in **Fig. 2.** If water vapor behaved as a perfect gas, its specific gravity referred to air as defined above would be constant, and equal to the ratio of molecular weights of water vapor and air.

$$G=\frac{M_w}{M_a}=\frac{18.016}{28.966}=0.6220$$

It will be noted that the value of G approaches the value of this ratio at the lower temperatures.

1.3—Moist Air Properties

To get a clear picture of the properties of moist air, we must first understand exactly what moist air is. Suppose one pound of air absolutely devoid of any water content is placed in a container in which the pressure may be maintained constant at atmospheric pressure. If a quantity of liquid water is now added to the container, a portion of it will evaporate and become mixed with the air in the form of water vapor. The remainder will continue to exist as a liquid in the bottom of the container. The process of evaporation requires the addition of a certain quantity of heat which, if the container is uninsulated, will be supplied by the surroundings. In this case, the temperature of the surroundings will be assumed to be 50 F.

If the container remains long enough at 50 F, an equilibrium condition will be established in which the air, water vapor, and liquid water will all be at exactly the same temperature. When this condition is established, the air is said to be saturated with water vapor and the material in the container, exclusive of the liquid water, is called **saturated air.**

The theory of psychrometrics based on Dalton's Principle assumes that, under saturation conditions such as those described above,

(1) The total pressure, p, is equal to the sum of the water vapor pressure, p_g'', at dew-point temperature and the partial pressure of the dry air, p_a.

$$\begin{aligned} p_a+p_g''&=p \\ p_a&=p-p_g'' \end{aligned} \tag{3}$$

(2) The partial pressure of the water vapor is equal to the saturation pressure of water as found in the steam tables corresponding to 50 F.

(3) Both the air and water vapor fractions in the saturated air behave as though the other were not present. Consequently, the volume of the mixture per lb of dry air may be calculated from Equation (1.1), us-

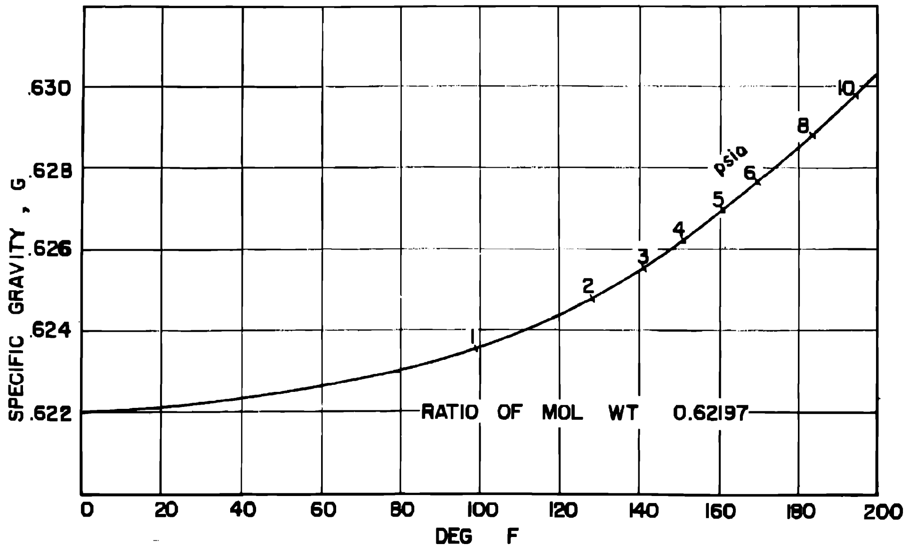


Fig. 2. Specific Gravity of Water Vapor Referred to Dry Air

ing the air partial pressure deduced from Equation (3).

$$v = \frac{R_a T}{p_a} = \frac{0.754307T}{p - p_g''} \tag{4}$$

Where v = volume of moist air, lb per lb dry air
R_a = gas constant for air, 0.75430
T = absolute temperature, $t +$ 459.7 (deg F abs)

(4) The enthalpy of the air-water vapor mixture is equal to the sum of the specific enthalpy of each constituent in Btu per lb, multiplied by the pounds of the constituents present in the mixture.

$$\begin{aligned} h &= h_a + W h_g \\ h - h_a &= W h_g \end{aligned} \tag{5}$$

Where h = enthalpy of moist air (Btu per lb dry air)
h_a = enthalpy of dry air (Btu per lb)
h_g = enthalpy of saturated water vapor at dry bulb temp, t (Btu per lb dry air)
W = specific humidity of air (lb per lb dry air).

Returning to the container with its saturated air and liquid water at 50 F, suppose the water is removed and the remaining moist air is heated at atmospheric pressure to 80 F. Obviously, the specific humidity, W, remains constant. It will be seen later that the water vapor pressure also remains substantially constant, although it does increase slightly. The condition of the water vapor is therefore changed and, corresponding to a pressure slightly above the vapor pressure at 50 F, is at a temperature of 80 F. It has become superheated steam. The air in this condition is said to be unsaturated because more water vapor could be added until the vapor pressure increased to that of water at 80 F.

If, instead of heating the 50 F saturated air, an attempt is made to cool it below 50 F, the actual vapor pressure will tend to exceed that corresponding to the lower temperature and condensation will occur. It is clear, then, that the temperature at which air of a certain moisture content becomes saturated upon cooling at constant pressure is the temperature at which condensation will begin. This temperature is called the **dew point**.

To avoid confusion, it must be noted that the moisture content, enthalpy and volume of moist air are always given **per lb of dry air in the mixture.** This practice, which deviates from that used in most other fields (wherein such properties are usually given **per lb of mixture**), is used because in psychrometric problems the quantity of water vapor continually varies while that of the dry air remains unchanged. The moisture content associated with 1 lb of dry air is called **specific humidity.**

Specific humidity, W. If the water vapor and dry air in a moist air sample are considered as occupying the same volume, the **moisture content**, usually called specific humidity, in lb of water vapor per lb of dry air, is defined by the ratio of the water vapor (ρ_w), and **air densities** (ρ_a) computed at their respective partial pressures. Since **specific volume** is the reciprocal of density, we may write

$$W = \frac{\rho_w}{\rho_a} = \frac{v_a}{v_w} \tag{6}$$

From Equation (1) the specific volume of air at a given temperature is inversely proportional to pressure. Using Equation (2), the above equation may therefore be written for the dew-point condition

$$W = \left(\frac{v_a}{v_g}\right)_{p_g'', t''} \times \frac{p_g''}{p - p_g''} = G \frac{p_g''}{p - p_g''} \tag{7}$$

Where W = specific humidity, lb of water vapor per lb of dry air
p_g'' = vapor pressure of water at dew-point temperature, t''
p = total or barometric pressure
v_a = specific volume of dry air
v_g = specific volume of water vapor at partial pressure p_g'' and absolute dew-point temperature t''.

Volume. By the assumption that in the mixture of water vapor and air, each constituent behaves as though the other were not present, it is clear that the net effect of adding water vapor to one pound of dry air is simply the reduction of the air pressure below the total pressure, p, by the amount of the water vapor pressure,

$$v = v_a \times \frac{p}{p - p_g''} \qquad (8)$$

$$v - v_a = v_a\left(\frac{p}{p - p_g''} - 1\right)$$

$$= v_a \frac{p_g''}{p - p_g''} = v_a \frac{W}{G} \qquad (9)$$

Where v = volume of moist air, cu ft per lb dry air

v_a = specific volume of dry air at the same temperature.

The last expression in Equation (9) is obtained by substituting from Equation (7) the ratio of specific humidity, W, to specific gravity of water vapor, G (at the dew-point temperature).

Enthalpy, h. It is obvious that Equation (5) represents the enthalpy of saturated air in Btu per lb of dry air. It is not so obvious, however, that this same equation also satisfactorily defines the enthalpy of unsaturated air, provided h_g is taken as the enthalpy of saturated water vapor at the dry-bulb temperature. To be exact, it would be necessary to use a value of enthalpy for the water vapor corresponding to its superheated condition, but at the relatively low vapor pressures encountered in moist air the enthalpy of superheated steam is almost exactly equal to that of saturated steam at the same temperature. Even at the upper limit of the psychrometric tables (200 F), a maximum error of less than 0.2% would occur from this approximation in the derived value of enthalpy for unsaturated air. The enthalpy of dry air and of saturated air would, of course, be more exact.

Relative humidity and saturation ratio. These two properties of air are defined by the following equations:

Relative humidity, $\phi = p_g''/p_g \qquad (10)$

Saturation ratio, $\mu = W/W_s \qquad (11)$

Where p_g'' = vapor pressure of water at dew-point temperature (in. Hg or psia)

p_g = vapor pressure of water at dry-bulb temperature (in. Hg or psia)

W = specific humidity, of air (lb per lb dry air)

W_s = specific humidity of saturated air at dry-bulb temperature (lb per lb dry air).

It is easy to show that, although these two properties are nearly equal at the lower end of the temperature range (say below 100 F), they are not actually identical. Writing Equation (7) for saturated and unsaturated air at the same temperature and total pressure, p, it is found that

$$\mu = \frac{W}{W_s} = \frac{p_g''}{p - p_g''} \bigg/ \frac{p_g}{p - p_g}$$

$$= \frac{p_g''}{p_g} \times \frac{p - p_g}{p - p_g''} = \phi \times \frac{p - p_g}{p - p_g''} \qquad (12)$$

Where p = total or atmospheric pressure.

Other symbols have the same significance as in Equations (10) and (11) above.

Moist air properties tables. The psychrometric tables in Chapter 10 are for atmospheric pressure but they include all the properties of water necessary for the computation of moist air properties at pressures higher or lower than atmospheric pressure. In the tables W_s = specific humidity of saturated air in grains per pound of dry air (1 lb = 7000 grains)

v_a = specific volume of dry air (cu ft per lb)

v_s = volume of saturated air (cu ft per lb of dry air)

h_a = enthalpy of dry air (Btu per lb)

h_s = enthalpy of saturated air (Btu per lb dry air).

It is obvious from inspection of Equations (5) and (9) that the increase in enthalpy due to the addition of water vapor is directly proportional to the specific humidity, W, and the increase in volume is practically so. This enables the determination of the volume or enthalpy of unsaturated air by a process of interpolation, using values of volume and enthalpy given in the psychrometric tables.

Thus

$$h = h_a + \mu(h_s - h_a) = h_s - (1-\mu)(h_s - h_a) \quad (13)$$

$$v = v_a + \mu(v_s - v_a) = v_s - (1-\mu)(v_s - v_a) \quad (14)$$

Interaction effects. It has already been noted that the individual properties of dry air and water vapor deviate slightly from the simple gas laws. Similarly, there is some evidence that the combination represented by moist air does not strictly follow the dictates of Dalton's Principle. Interactive molecular forces, and the slight solubility of air in any liquid water in contact with saturated air, result in a very small increase in moisture content of saturated air and other related properties. Goff[3] has considered these factors in the preparation of his psychrometric data for standard atmospheric pressure. **Table 1** shows a comparison of his values with values from the tables in Chapter 10, which are based on the simpler psychrometric theory outlined above and ignore the interaction and other secondary effects. Little or no information exists on the effect of pressure on these phenomena.

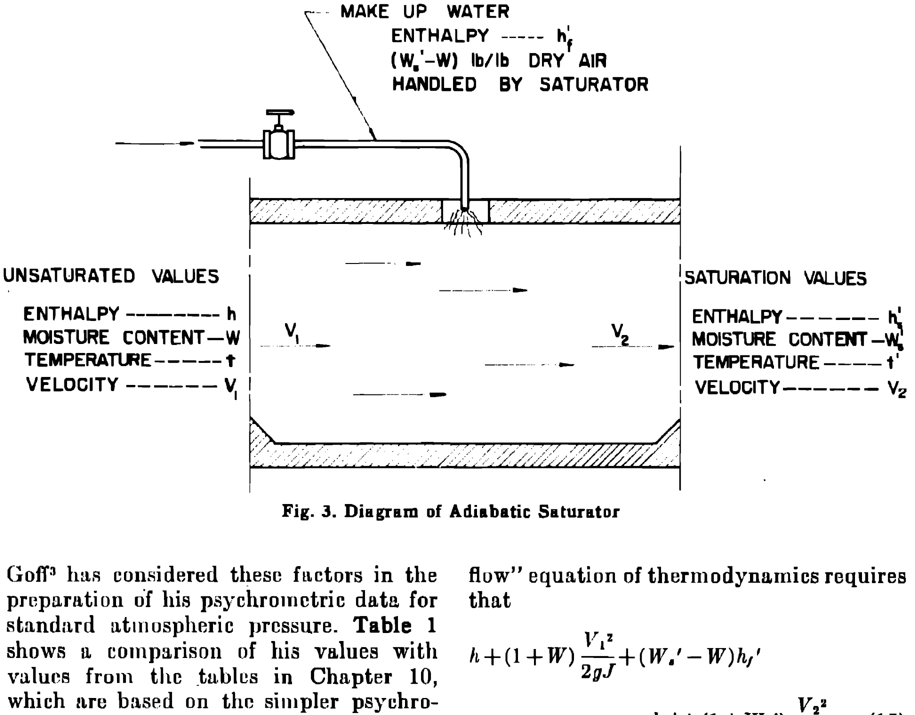


Fig. 3. Diagram of Adiabatic Saturator

Evaporative cooling and thermodynamic wet bulb. Referring to **Fig. 3,** if a stream of unsaturated air having temperature *t*, enthalpy *h*, and specific humidity *W*, is passed through a perfectly insulated chamber where it comes in contact with large wetted surfaces, it will leave in a substantially saturated condition with temperature *t'*, enthalpy h_s' and moisture content W_s'. If the process is continous, a quantity of liquid water $(W_s' - W)$ must be supplied for each pound of air passing through the saturator. Now, let it be assumed that:

1. There is no pressure drop in the process.
2. The temperature of the water added equals *t'* and its kinetic energy is zero.
3. There is no mechanical work or heat added during the saturation process.

Under such conditions, the "steady flow" equation of thermodynamics requires that

$$h + (1+W)\frac{V_1^2}{2gJ} + (W_s' - W)h_f' = h_s' + (1+W_s')\frac{V_2^2}{2gJ} \quad (15)$$

The superscripts indicate values at the thermodynamic wet-bulb temperature, *t'*, defined below.

Where, in addition to the symbols described above,

V = velocity, fps
g = acceleration of gravity, 32.2 fps^2

Table 1. Saturation Properties

	Neglecting interaction effects (Section 10)			Including interaction effects[1]		
Temp, F	W_s	h_s	v_s	W_s	h_s	v_s
−50	.2893	−11.97	10.31	.2914	−11.969	10.314
−60	.1471	−14.40	10.06	.1483	−14.394	10.059
Diff*	.1422	2.43		.1431	2.425	
50	53.40	20.28	13.00	53.61	20.301	13.001
40	36.34	15.21	12.70	36.49	15.230	12.695
Diff*	17.06	5.07		17.12	5.071	
200	16052	2675.6	77.14	16065	2677	77.142
190	7682	1298.9	44.94	7693	1301	44.959
Diff*	8370	1376.7		8372	1376	

* Differences in specific humidity (W) and enthalpy (h) for a 10 F temperature range, since comparison of absolute values has little practical significance.

J = mechanical equivalent of heat (778 ft-lb per Btu)

If the velocity terms are neglected, this is reduced to

$$h+(W_s'-W)h_f'=h_s' \quad \text{or} \qquad (16)$$
$$h-Wh_f'=h_s'-W_s'h_f'$$

It will be noted that the enthalpy of the saturated air, h_s', its specific humidity, W_s', and the enthalpy of liquid water, h_f', are all defined by the leaving air temperature, t'. Therefore, for a given t'

$$h_s'-W_s'h_f'=\Sigma$$

where this equation defines the term Σ known as the Sigma function.

Obviously, there are any number of unique moist air samples which will satisfy Equation (16), ranging from dry air, whose enthalpy $=h_s'$, to saturated air at t'. All such samples are said to have the same thermodynamic wet-bulb temperature.

Equation (16), which defines the **thermodynamic wet bulb** (twb) and the process of evaporative cooling which it describes, are important because both are very closely representative of the conditions in the film of air surrounding a wet-bulb thermometer (thermometer with bulb covered by a wetted wick). In fact, it is possible to calibrate, either theoretically or experimentally, the readings of a wet-bulb thermometer so that the twb may be obtained. Given the twb and the dry-bulb temperature, Equation (16) may be solved for W and all the related psychrometric properties may be determined.

Hence it may be said that the **process of evaporative cooling** to the twb is the fundamental principle on which the wet-dry bulb psychrometer is based.

The velocity terms of Equation (15) were neglected in the development of the twb concept. There is no objection to this procedure under most circumstances. At higher velocities, of the order of 70 fps and above and for precise laboratory work, the magnitude of the velocity terms in Equation (15) should be investigated. If the kinetic energy entering the saturator is dissipated in friction, the pressure remaining constant, the resulting rise in twb is less than the corresponding rise of db. The wet-bulb depression is consequently greater. On the other hand, if the kinetic energy is reduced to zero by compression, a somewhat higher twb rise results and the wb depression deviates less from that determined by neglecting the velocity terms.

II. The Psychrometric Chart

2.1—Description of the Charts

The psychrometric chart is the most convenient means for determining the **properties of moist air.** It is particularly useful for representing processes in which air is cooled, heated, humidified, dehumidified, or mixed with air from other sources.

The charts used here contain an outstanding improvement over charts formerly available. Enthalpy of unsaturated air has previously been difficult to show on the psychrometric chart, because the constant enthalpy lines are so nearly parallel to the wet-bulb lines that both cannot be shown without causing confusion. Thus some previous charts have shown enthalpy scales that were only approximately correct, tending to large errors at high temperatures. Other charts used the "Sigma heat function" which, by definition, required a correction for the enthalpy of the moisture contained in the air. In the present charts the enthalpy is given on a scale of enthalpy at saturation. The readings from this scale are used in combination with a set of deviation contours showing the **deviation of enthalpy** for conditions other than saturation. This combination gives exact values, or when desired the deviation lines may be ignored with results that will be sufficiently correct for most problems in the comfort air conditioning range. The new type of chart is particularly valuable at temperatures below 32 F and at high temperatures above comfort conditions. In both of these ranges, the deviation of enthalpy for conditions other than saturation cannot be ignored without causing large errors.

The following properties can be determined from the charts, as illustrated in **Fig. 4**:

Dry bulb temperature, represented by vertical lines, is read along the lower edge of the chart.

Wet bulb temperature, represented by diagonal lines, is read at the intersection of these lines with the saturation curve. Below 32 F the charts show properties of moist air based upon the vapor pressure over ice.

Specific humidity in grains or pounds per pound of dry air, represented by horizontal lines, is read on the scales at the right-hand side of the chart.

Dew point (saturation) temperatures, constant along horizontal lines parallel to the specific humidity lines, are read along the saturation curve. Note that dry-bulb, wet-bulb and dew-point temperatures coincide at the saturation curve.

Percent relative humidity is represented by a family of curved lines with values indicated thereon. Interpolation between lines of relative humidity is done along vertical or dry-bulb lines, since spacing is approximately uniform in that direction.

Volume per pound of dry air is indicated by diagonal lines with values marked thereon.

Enthalpy at saturation is read by extending the wet-bulb lines to intersection with the diagonal scale.

Enthalpy deviation for conditions other than saturation is indicated by curved, dashed lines with values marked thereon. Note that below 32 F wet bulb these lines are nearly parallel to the saturation line and are much closer together, indicating a greater deviation. Interpolation between enthalpy deviation lines should be done along wet-bulb lines, since spacing is uniform in that direction.

Enthalpy of water or ice added to or rejected from the system is read on small diagrams included with the main chart. The enthalpy of the rejected or added water constitutes a small but definite part of the heat balance. For the approximate solution of comfort air conditioning problems, it is often neglected, and the error from this source seldom exceeds 1%. The error is much greater, however, at high temperatures and at temperatures below freezing. The enthalpy of the rejected or added water can be conveniently read from the small diagrams. In order to use these diagrams, simply determine the difference in moisture between the entering and leaving state, determine the temperature at which the water leaves or enters the system, and read its enthalpy directly from the small chart. Note that where the moisture freezes on the coil, a much greater enthalpy is involved and it is negative, so that it becomes an addition to the cooling load.

Sensible heat factor. In the application of air conditioning equipment, certain methods of calculation involve the ratio of the sensible heat to the total heat load. This ratio is approximately defined as the change in enthalpy which takes place between the two dry-bulb temperatures at the mean moisture content to the total enthalpy difference. A scale along the right-hand margin of the normal temperature chart, together with an origin at 80 db and 50% relative humidity, affords a convenient means of reading the sensible heat factor.

Specific properties. It has been found convenient in practice to represent the enthalpy, volume and moisture content of air on the basis of the weight of dry air or **per pound of dry air.** The weight of mixture is obtained by simply adding the weight of the moisture or by multiplying the weight of dry air by $(1+W)$, where W equals the pounds of moisture (i.e., specific humidity) per pound of dry air. Simularly, the volume occupied by one pound of mixture is obtained by dividing the volume per pound of dry air by $(1+W)$.

Air is generally measured and referred to on a cubic foot basis and, therefore, it is often necessary to determine moisture content and heat content per cubic foot. This is conveniently accomplished by dividing the moisture content or enthalpy per pound of dry air by the volume per pound of dry air. The result will be moisture content or enthalpy per cubic foot of mixture.

Barometric pressure. The relative proportion of moisture in the air for any particular set of wet-bulb and dry-bulb temperatures depends upon the total or barometric pressure. The psychrometric chart represents the properties of air for standard barometric pressure of 29.92 in. of mercury. A convenient table is provided for extending these properties to other barometric pressures. For psychrometric problems in the comfort air conditioning range, where the highest accuracy is not required and the correction for barometric pressure does not exceed 1 in. of mercury, it is sometimes ignored.

Symbols Used in Text and Examples

t = dry bulb temperature, F

t' = wet bulb temperature, F

t'' = dew point temperature, F

W = specific humidity, lb of water vapor per lb of dry air or grains of water vapor per lb of dry air (7,000 grains = 1 lb)

W_s = Specific humidity of air saturated with water vapor at the dry bulb temperature, lb per lb of dry air or grains per lb of dry air

T = absolute temperature, equals (F temperature + 459.7)

p = total or barometric pressure, inches of mercury (in. Hg)

p_a = partial pressure of air, in. Hg

p_g'' = pressure of saturated water vapor at the dew-point temperature, in. Hg

p_g = pressure of saturated water vapor at the dry-bulb temperature, in. Hg

v = volume, cu ft per 1 lb of dry air with its associated water vapor

v_a = sp volume of air; volume in cu ft occupied by 1 lb of dry air at pressure p_a and temperature t

v_w = volume in cu ft occupied by 1 lb of water vapor at its partial pressure p_w and temperature t

ρ_a = density $= 1/v_a$

ρ_w = density $= 1/v_w$

ϕ = relative humidity, p_g''/p_g

μ = saturation ratio (also known as percentage humidity) $= W/W_s$

h = enthalpy of moist air, Btu per lb of dry air (and associated water vapor)

h_s' = enthalpy of air saturated at the wet-bulb temperature, Btu per lb dry air

h_a = enthalpy of dry air, Btu per lb

h_g = enthalpy of saturated water vapor at dry-bulb temperature, Btu per lb

h_m = enthalpy change from water added or rejected from system, in Btu per lb of dry air

d = enthalpy deviation, Btu per lb of dry air

$h = h_s' + d$ = enthalpy (true) of moist air as found from charts, Btu per lb of dry air

m = Weight of moisture added to or rejected from air stream per lb of dry air, in grains or lb

q = Heat added to or removed from system, Btu per lb of dry air

Subscripts 1, 2, 3 etc. indicate entering and progressive state points in the system (t_1, t_2, t_3, etc.)

2.2—Examples

Example 1. Properties of Moist Air (see Fig. 4). Find the properties of moist air when the dry bulb is 80 F and the wet bulb is 67 F.

Solution (read directly from psychrometric chart for normal temperatures)

Specific humidity (moisture content) = (W) = 78.2 gr per lb of dry air
Enthalpy at saturation (h_s') = 31.62 Btu per lb of dry air
Enthalpy deviation (d) = −0.1 Btu per lb of dry air
True enthalpy (h) = 31.52 Btu per lb of dry air
Specific volume (v) = 13.8 cu ft per lb of dry air
Relative humidity (ϕ) = 51%
Dew point (t'') = 60.3 F

Example 2. Heating Process (see **Fig. 5**). Air is heated by a steam coil from 30 F dry bulb and 80% relative humidity to 75 F dry bulb. Find the relative humidity, wet-bulb temperature, and dew point of the heated air.

Solution (from normal-temperatures psychrometric chart)

(t'') Dew point = 25.2 F
(t_2') Wet-bulb temperature of heated air = 51.5 F
(ϕ_2) Relative humidity = 15%

Example 2a. Find the heat added per lb of dry air in Example (2).

Approximate Solution

$h_{s1}' = 10.1$
$h_{s2}' = 21.1$
$q = h_{s2}' - h_{s1}' = 11$ Btu per lb of dry air

(Note that this result is 1.5% higher than the following exact solution.)

Exact Solution

$h_1 = h_{s1}' + d_1 = 10.1 + .06 = 10.16$ Btu
$h_2 = h_{s2}' + d_2 = 21.1 - 0.1 = 21.0$ Btu
$q = h_2 - h_1 = 10.84$ Btu

Example 3. Spray or Evaporative Cooler (see **Fig. 6**). Air at 95 db and 70 wb passes through a water spray where its relative humidity is increased to 90%. The spray water is recirculated and the make-up water enters at 70 F. Determine the leaving dry-bulb temperature, wet-bulb temperature, change in enthalpy of the air, and the amount of moisture added per lb of dry air.

Solution. Since the make-up water enters at the wet-bulb temperature of 70 F and no heat is added to or removed from the system, this is by definition an adiabatic process. There will be no change in the wet-bulb temperature and the leaving dry-bulb temperature can, therefore, be determined from the chart, by following the 70 F wet-bulb line to the 90% relative humidity line, to be 72.2 F.

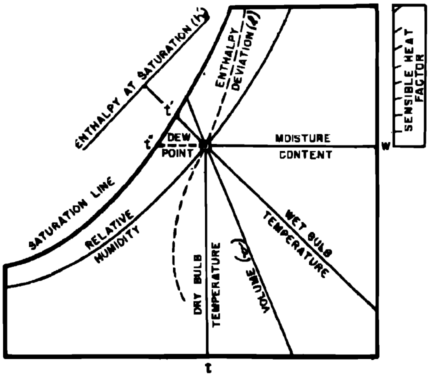


Fig. 4. Diagram of Psychrometric Chart Showing Properties of Moist Air

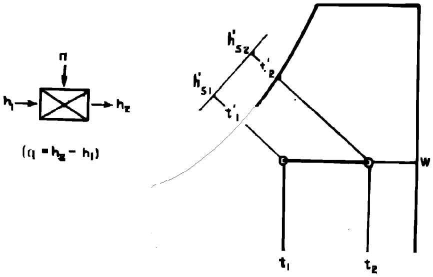


Fig. 5. Heating Process

The only change of enthalpy of the air is that due to the enthalpy of the make-up water. This can be demonstrated as follows:

$W_1 = 70$ gr per lb of dry air
$W_2 = 107$ gr per lb of dry air
$m = W_2 - W_1 = 37$ gr per lb of dry air
$h_1 = h_{s1}' + d_1 = 34.1 - 0.22 = 33.88$ Btu
$h_2 = h_{s2}' + d_2 = 34.1 - 0.02 = 34.08$ Btu
h_m = (from small diagram for 37 gr at 70 F) = 0.2 Btu
$q = h_1 - h_2 + h_m = 33.88 - 34.08 + 0.2 = 0$

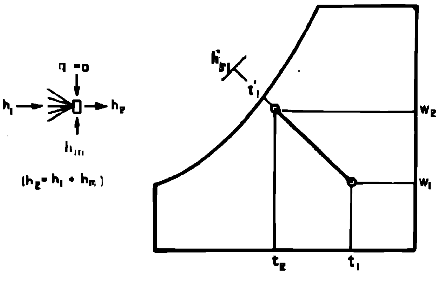


Fig. 6. Spray or Evaporative Cooler

Example 4. Cooling and Dehumidifying Process. (see **Fig. 7**). Find the cooling effect per pound of dry air where the air enters a cooling coil at 83 db, 69 wb and leaves at 56 db, 55 wb and the condensate is rejected at 55 F.

Approximate Solution

$h_{s1}' = 33.24$ Btu
$h_{s2}' = 23.21$ Btu
$q = h_{s1}' - h_{s2}' = 10.03$ Btu

(Note that this value is 1.9% greater than the following exact solution.)

Exact Solution

$h_1 = h_{s1}' + d_1 = 33.24 - 0.12 = 33.12$ Btu
$h_2 = h_{s2}' + d_2 = 23.21 - 0.01 = 23.20$ Btu
$m = W_1 - W = 84 - 63 = 21$ gr
$h_m =$ (from small diagram, 21 gr at 55 F) $= 0.08$
$q = h_1 - h_2 - h_m = 33.12 - 23.20 - 0.08 = 9.84$

Example 4a. For the conditions of Example (4) determine the sensible heat factor.

Solution. As shown in Fig. 7, draw a line between the entering and leaving conditions. Then extend a parallel line from the origin located at 80 db and 50% relative humidity to the sensible heat factor scale and read the sensible heat factor of 0.68.

Example 4b (see Fig. 7). Find the cooling load per pound of dry air due to infiltration of room air at 80 db, 67 wb into a cooler maintained at 30 db, 28 wb, where moisture freezes on the coil which is maintained at 20 F.

Approximate Solution

$h_{s1}' = 31.62$ Btu
$h_{s2}' = 10.10$ Btu
$q = h_{s1}' - h_{s2}' = 21.52$ Btu

(Note that this result is 5% lower than the following exact solution.)

Exact Solution

$h_1 = h_{s1}' + d_1 = 31.62 - .1 = 31.52$ Btu
$h_2 = h_{s2}' + d_2 = 10.10 + .06 = 10.16$ Btu
$m = W_1 - W_2 = 78 - 19 = 59$ gr
$h_i =$ (from diagram, 59 gr, 20 F) $= -1.26$ Btu
$q = h_1 - h_2 - h_i = 31.52 - 10.16 + 1.26 = 22.62$

Example 5. Cooling Tower (see **Fig. 8**). Determine the water consumption and the amount of heat dissipated per 1,000 cu ft of entering air at 90 db, 70 wb, when the air leaves at 110 F saturated and make-up water is at 75 F.

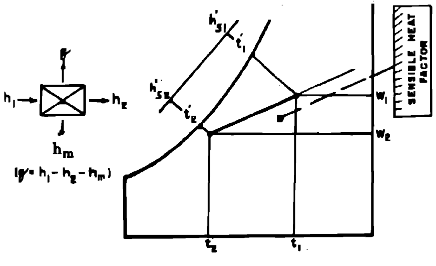


Fig. 7. Cooling and Dehumidifying Process

Approximation Solution (high temperature chart)

$m = W_2 - W_1 = 416 - 78 = 338$ gr added
$q = h_{s2}' - h_{s1}' = 92.34 - 34.1 = 58.24$ Btu added
Specific volume of entering air = 14.1 cu ft per lb of dry air

$$\text{Water consumption per 1,000 cfm} = \frac{338 \times 1000}{7{,}000 \times 14.1} = 3.43 \text{ lb}$$

$$\text{Heat dissipated per 1,000 cfm} = \frac{58.24 \times 1{,}000}{14.1} = 4{,}130 \text{ Btu}$$

(Note that this result is 3.4% greater than the following exact solution.)

Exact Solution

$m = W_2 - W_1 = 416 - 78 = 338$ gr
$h_m = 2.08$ Btu (from diagram on high temperature chart, 338 gr, 75 F)
$h_1 = h_{s1}' + d_1 = 34.1 - 0.18 = 33.92$
$h_2 = h_{s2}' + d_2 = 92.34 + 0 = 92.34$
$q = h_2 - h_1 - h_m = 92.34 - 33.92 - 2.08 = 56.34$
Specific volume of entering air = 14.1 cu ft per lb of dry air

$$\text{Heat dissipated per 1,000 cfm} = \frac{56.34 \times 1{,}000}{14.1} = 3{,}990$$

Water consumption will be the same as shown above.

Example 6. Mixture of Air (see **Fig. 9**). Inside air at 75 db and 62 wb is mixed with outside air at 95 db and 75 wb in the proportion of one part outside air to three parts inside air by weight. Find the properties of the resulting mixture.

Solution. Draw a straight line between the inside and outside state points. Measure off one-fourth of the length of this line, starting from the inside state. This will establish a point representing the mixture of one part outside air with three parts inside air. Properties of the mixed air are then determined as explained in Example (1).

It is often convenient to use the dry-dulb temperature scale or the specific humidity scale as the means of dividing the mixture line into the desired proportions. In this example, for instance, one-fourth of the difference between the inside and outside dry-bulb temperatures is 5 F, and thus the mixture of one part outside air with three parts inside air will be at a temperature of 75+5=80 F. At the intersection of 80 F dry-bulb temperature and the line representing the mixture, read the specific humidity of 71.3 gr, wet bulb of 65.6 F and enthalpy of 30.6 Btu.

When the two air quantities being mixed are at widely different temperatures, there will be a very

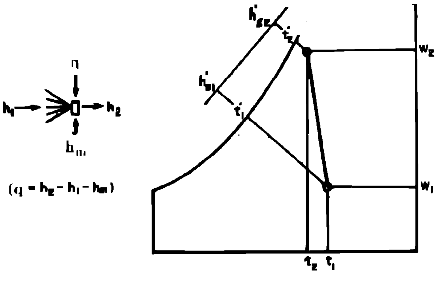


Fig. 8. Cooling Tower

small error in the above method due to the slight variation of the specific heat of moist air. When an exact solution is required under such conditions as laboratory work, it is recommended that the mixture be calculated on the basis of specific humidity and enthalpy.

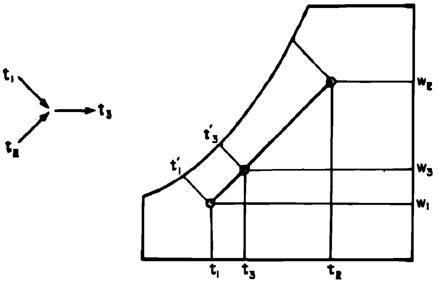


Fig. 9. Mixture of Air

Use of the Chart at Barometric Pressures other than 29.92 in. Mercury

Each chart is provided with a table showing the change of specific humidity at saturation, $\Delta W_s'$, and change of enthalpy, Δh, for various barometric pressures. True values of **specific humidity** (W) and **enthalpy** (h) are obtained by adding the corrections corresponding to the actual barometer reading to the values taken from the chart.

The specific **volume** of moist air in cu ft per lb of dry air can be determined for any barometric pressures by use of the formula:

$$v=\frac{0.754(t+460)}{p}\left(1+\frac{W}{4{,}360}\right)$$

Where: v = volume of moist air per lb of dry air, cu ft
t = dry-bulb temperature
p = barometric pressure, in. Hg
W = specific humidity in gr per lb of dry air corrected for the actual barometric pressure.

Relative humidity and **dew point** can be determined for other than standard barometric pressures by the use of a table of vapor pressures and a simple calculation. To determine either relative humidity or dew point, it is first necessary by use of the following formula to calculate the vapor pressure of the moisture that is present in the air.

$$p_g''=\frac{W\times p}{4{,}360+W}$$

Where p = barometric pressure
p_g'' = vapor pressure, in same units used for barometric pressure
W = specific humidity, in gr per lb of dry air

Dew point is then read directly from a table of vapor pressures as the temperature corresponding to the calculated vapor pressure.

Relative humidity is determined by dividing the vapor pressure as obtained above by the vapor pressure corresponding to saturation at the dry bulb temperature.

$$\text{Relative humidity}=\frac{p_g''}{p_g}$$

Where: p_g'' is the vapor pressure of the moisture in the air calculated as shown above.
p_g is the vapor pressure corresponding to saturation at the dry bulb temperature.

Note that the factor 4,360 is obtained by multiplying the specific weight of water vapor referred to air by 7,000, the number of grains per pound. The specific weight of water vapor, as shown in Fig. 2, varies slightly with temperature and for accurate results the above factors should be modified as follows for different temperature ranges:

Up to 90 F, use 4,360
90 F to 150 F, use 4,380
150 F to 200 F, use 4,400

Example 7. For a barometric pressure of 25.92 in. Hg ($\Delta p=-4$), and a reading of 90 F dry-bulb and 70 F wet bulb temperature, determine the following:

Specific humidity (W)
Enthalpy (h)
Dew point (t'')
Relative humidity (ϕ)
Volume per lb of dry air (v_a)

Solution. From the chart read specific humidity of 78 gr, corresponding to 90 db and 70 wb. From the table under $\Delta p=-4$ and $t'=70$, read $\Delta W_s'=17.6$ gr. This is the barometric correction for the saturated conditions, and as noted on the chart this correction should be reduced for conditions other than saturated in the ratio of 1% for each 24 F of wet bulb depression. Thus

$$\Delta W=17.6\left(1-\frac{20}{24}\times .01\right)=17.5$$

and $W=78+17.5=95.5$.

Enthalpy at saturation of 34.10 Btu is read on the diagonal scale of the chart for 70 wb and corrected for the deviation of 0.18 at 90 db, giving the value of $34.10-0.28=33.92$. To this must be added the correction of 2.75 read from the table for $\Delta p=-4$ and $t'=70$, giving the true enthalpy of $33.92+2.75=36.67$.

To determine dew point first calculate the vapor pressure as follows:

$$p_g''=\frac{W\times p}{4360+W}=\frac{95.5\times 25.92}{4360+95.5}=.555\text{ in. Hg}$$

Then from the table of vapor pressures determine the dew point temperature of 61.4 F.

Relative humidity is determined by dividing the

PSYCHROMETRIC CHART
LOW TEMPERATURES
Barometric Pressure 29.92 in. Hg

ADDITIVE CORRECTIONS FOR W AND h WHEN BAROMETRIC PRESSURE DIFFERS FROM STANDARD BAROMETER

Wet Bulb Temp. t'	Sat. Vapor Press. in. Hg	APPROXIMATE ALTITUDE IN FEET													
		−900		900		1800		2700		3700		4600		5600	
		Δp = +1		Δp = −1		Δp = −2		Δp = −3		Δp = −4		Δp = −5		Δp = −6	
		ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh
−20	0.013	−0.06	−0.01	0.06	0.01	0.12	0.02	0.18	0.03	0.28	0.04	0.36	0.05	0.47	0.07
−18	0.014	−0.07	−0.01	0.07	0.01	0.14	0.02	0.21	0.03	0.32	0.05	0.41	0.06	0.53	0.08
−16	0.016	−0.07	−0.01	0.08	0.01	0.16	0.02	0.24	0.04	0.36	0.06	0.46	0.07	0.59	0.09
−14	0.018	−0.08	−0.01	0.09	0.01	0.18	0.03	0.27	0.04	0.40	0.06	0.52	0.08	0.65	0.10
−12	0.020	−0.09	−0.01	0.10	0.01	0.21	0.03	0.31	0.05	0.44	0.07	0.58	0.09	0.72	0.11
−10	0.022	−0.10	−0.02	0.11	0.02	0.23	0.03	0.36	0.05	0.50	0.07	0.64	0.10	0.81	0.12
−8	0.025	−0.12	−0.02	0.13	0.02	0.26	0.04	0.40	0.06	0.55	0.08	0.72	0.11	0.90	0.13
−6	0.027	−0.13	−0.02	0.14	0.02	0.29	0.04	0.44	0.07	0.62	0.09	0.80	0.12	1.00	0.15
−4	0.030	−0.14	−0.02	0.15	0.02	0.32	0.05	0.50	0.07	0.69	0.10	0.89	0.13	1.12	0.17
−2	0.034	−0.16	−0.02	0.17	0.02	0.35	0.05	0.55	0.08	0.76	0.11	0.99	0.15	1.24	0.19
0	0.038	−0.18	−0.03	0.19	0.03	0.39	0.06	0.61	0.09	0.85	0.13	1.10	0.17	1.38	0.21
2	0.042	−0.20	−0.03	0.21	0.03	0.44	0.07	0.68	0.10	0.94	0.14	1.22	0.19	1.53	0.23
4	0.046	−0.22	−0.03	0.23	0.03	0.48	0.07	0.75	0.11	1.05	0.16	1.36	0.21	1.70	0.26
6	0.051	−0.24	−0.04	0.26	0.04	0.54	0.08	0.83	0.13	1.16	0.18	1.51	0.23	1.89	0.29
8	0.057	−0.27	−0.04	0.29	0.04	0.59	0.09	0.93	0.14	1.29	0.19	1.67	0.25	2.09	0.32
10	0.063	−0.30	−0.04	0.32	0.05	0.66	0.10	1.03	0.16	1.42	0.22	1.85	0.28	2.31	0.35
12	0.069	−0.33	−0.05	0.35	0.05	0.73	0.11	1.13	0.17	1.57	0.24	2.04	0.31	2.56	0.39
14	0.077	−0.36	−0.05	0.39	0.06	0.81	0.12	1.25	0.19	1.74	0.26	2.26	0.34	2.83	0.43
16	0.085	−0.40	−0.06	0.43	0.06	0.89	0.14	1.38	0.21	1.92	0.29	2.49	0.38	3.12	0.48
18	0.093	−0.44	−0.07	0.47	0.07	0.98	0.15	1.53	0.23	2.12	0.32	2.75	0.42	3.44	0.53
20	0.103	−0.49	−0.08	0.53	0.08	1.08	0.17	1.66	0.26	2.33	0.36	3.03	0.46	3.79	0.58
22	0.113	−0.5	−0.08	0.6	0.09	1.2	0.18	1.8	0.29	2.6	0.40	3.4	0.52	4.2	0.64
24	0.124	−0.6	−0.09	0.6	0.10	1.3	0.20	2.1	0.32	2.8	0.43	3.7	0.57	4.6	0.71
26	0.137	−0.7	−0.10	0.7	0.11	1.4	0.22	2.3	0.35	3.1	0.48	4.1	0.63	5.1	0.78
28	0.150	−0.7	−0.11	0.8	0.12	1.6	0.24	2.5	0.38	3.4	0.52	4.5	0.69	5.6	0.86
30	0.165	−0.8	−0.12	0.8	0.13	1.7	0.27	2.7	0.42	3.8	0.58	4.9	0.75	6.1	0.92
32	0.180	−0.9	−0.13	0.9	0.14	1.9	0.29	3.0	0.45	4.1	0.63	5.3	0.82	6.6	1.01
34	0.197	−0.9	−0.14	1.0	0.15	2.1	0.32	3.2	0.49	4.4	0.68	5.7	0.89	7.2	1.11
36	0.212	−1.0	−0.15	1.1	0.17	2.3	0.35	3.5	0.53	4.8	0.74	6.2	0.96	7.9	1.20
38	0.229	−1.1	−0.17	1.2	0.18	2.4	0.37	3.8	0.58	5.2	0.80	6.8	1.05	8.4	1.30
40	0.248	−1.2	−0.18	1.3	0.20	2.6	0.41	4.1	0.63	5.7	0.88	7.4	1.14	9.2	1.42
42	0.266	−1.3	−0.20	1.4	0.21	2.9	0.44	4.4	0.69	6.1	0.94	8.0	1.23	10.0	1.54
44	0.289	−1.4	−0.21	1.5	0.23	3.1	0.47	4.8	0.74	6.7	1.04	8.7	1.34	10.8	1.67
46	0.312	−1.5	−0.22	1.6	0.25	3.3	0.51	5.2	0.80	7.2	1.12	9.4	1.45	11.7	1.81
48	0.336	−1.6	−0.23	1.8	0.27	3.6	0.55	5.6	0.87	7.8	1.21	10.2	1.58	12.6	1.95

t = Dry bulb temperature (F).
t' = Wet bulb temperature (F).
p = Barometric pressure (in. of Hg).
Δp = Pressure difference from standard barometer (in. of Hg).
W = Specific humidity of air (gr per lb of dry air).
W's = Specific humidity of air saturated at wet bulb temperature t' (gr per lb of dry air).
ΔW = Specific humidity correction of air when barometric pressure differs from standard barometer (gr per lb of dry air).
ΔW's = Specific humidity correction of air saturated at wet bulb temperature when barometric pressure differs from standard barometer (gr per lb of dry air).

NOTE: To obtain ΔW reduce value of ΔW's by 1% where t − t' = 24 F and correct proportionally when t − t' is not 24 F.

h = Enthalpy of moist air (Btu per lb of dry air).
Δh = Enthalpy correction when barometer pressure differs from standard barometer, for saturated or unsaturated air (Btu per lb of dry air).
v = Volume of moist air (cu ft per lb of dry air).

$$v = \frac{.754\,(t + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$

PROPERTIES OF SATURATED AIR
−20 to −100 F

Temp. F	Vapor Press. in. Hg	Moisture Content Grains*	Volume cu ft*	Enthalpy Btu*
−20	.0126	1.83	11.08	−4.58
−22	.0112	1.63	11.03	−5.08
−24	.0100	1.45	10.98	−5.54
−26	.0089	1.29	10.93	−6.04
−28	.0079	1.15	10.88	−6.55
−30	.0070	1.02	10.83	−7.06
−32	.0062	0.90	10.78	−7.54
−34	.0055	0.80	10.73	−8.04
−36	.0049	0.71	10.68	−8.53
−38	.0043	0.63	10.63	−9.03
−40	.0038	0.55	10.58	−9.52
−42	.0033	0.49	10.53	−10.01
−44	.0029	0.43	10.48	−10.50
−46	.0026	0.38	10.43	−10.98
−48	.0023	0.33	10.38	−11.47
−50	.0020	0.29	10.33	−11.96
−52	.0017	0.25	10.28	−12.44
−54	.0015	0.22	10.23	−12.93
−56	.0013	0.19	10.18	−13.41
−58	.0012	0.17	10.13	−13.89
−60	.0010	0.15	10.08	−14.38
−62	.0009	0.13	10.03	−14.86
−64	.0008	0.11	9.98	−15.34
−66	.0007	0.10	9.93	−15.83
−68	.0006	0.08	9.87	−16.31
−70	.0005	0.07	9.82	−16.79
−72	.0004	0.06	9.77	−17.27
−74	.0004	0.05	9.72	−17.75
−76	.0003	0.05	9.67	−18.23
−78	.0003	0.04	9.62	−18.71
−80	.0002	0.03	9.57	−19.19
−82	.0002	0.03	9.52	−19.68
−84	.0002	0.03	9.47	−20.16
−86	.0001	0.02	9.42	−20.64
−88	.0001	0.02	9.37	−21.12
−90	.0001	0.02	9.32	−21.60
−92	.0001	0.01	9.27	−22.08
−94	.0001	0.01	9.22	−22.56
−96	.0001	0.01	9.17	−23.04
−98	.0001	0.01	9.12	−23.52
−100	.0000	0.01	9.07	−24.00

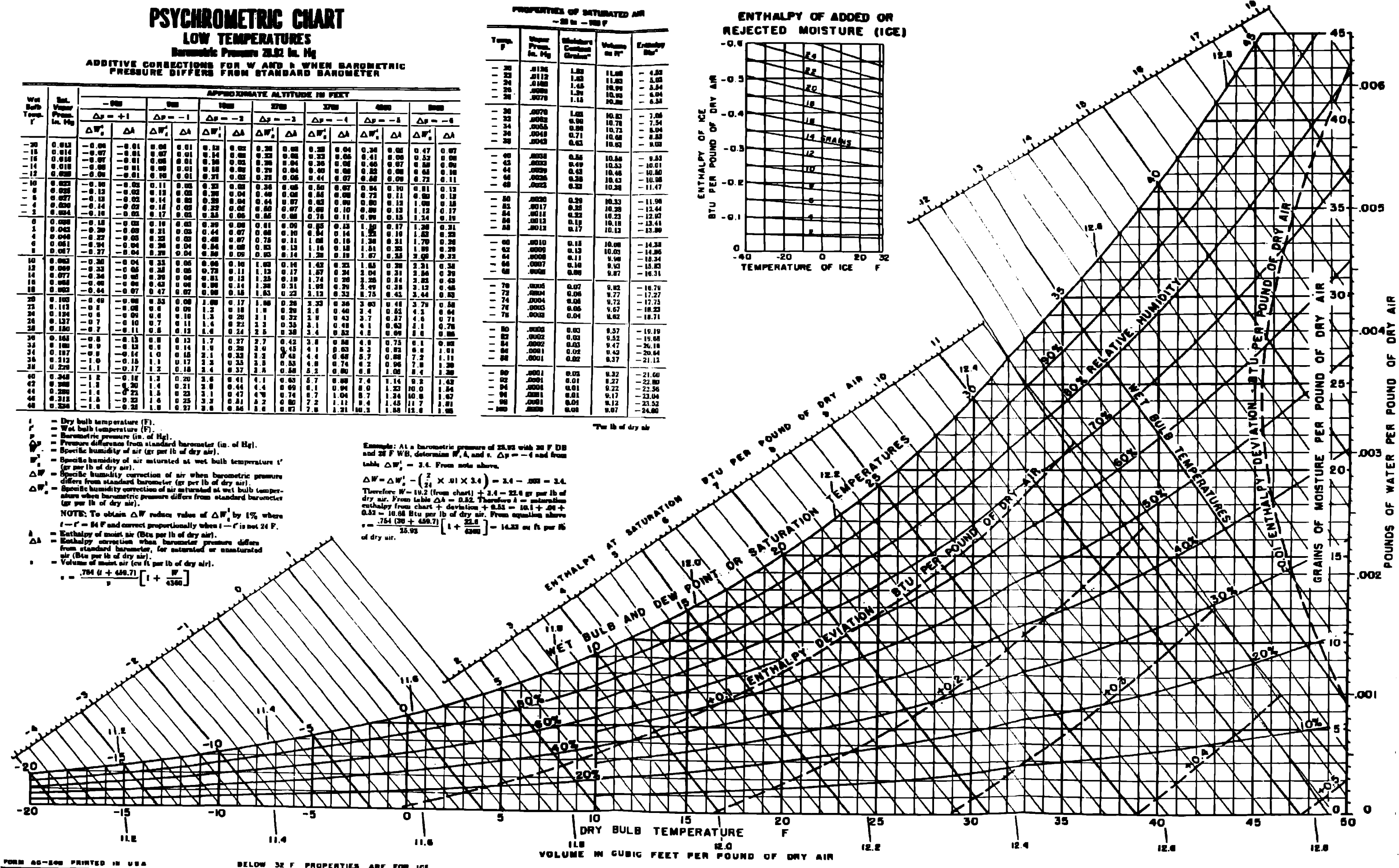
*Per lb of dry air

Example: At a barometric pressure of 25.93 with 30 F DB and 28 F WB, determine W, h, and v. Δp = −4 and from table ΔW's = 3.4. From note above,

$$\Delta W = \Delta W'_s - \left(\frac{2}{24} \times .01 \times 3.4\right) = 3.4 - .003 = 3.4.$$

Therefore W = 19.2 (from chart) + 3.4 = 22.6 gr per lb of dry air. From table Δh = 0.52. Therefore h = saturation enthalpy from chart + deviation + 0.52 = 10.1 + .06 + 0.52 = 10.68 Btu per lb of dry air. From equation above

$$v = \frac{.754\,(30 + 459.7)}{25.93}\left[1 + \frac{22.6}{4360}\right] = 14.33 \text{ cu ft per lb of dry air.}$$



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PSYCHROMETRIC CHART
NORMAL TEMPERATURES
Barometric Pressure 29.92 in. Hg

ADDITIVE CORRECTIONS FOR W AND h WHEN BAROMETRIC PRESSURE DIFFERS FROM STANDARD BAROMETER

Wet Bulb Temp. t'	Sat. Vapor Press. In. Hg	APPROXIMATE ALTITUDE IN FEET													
		−900		900		1800		2700		3700		4600		5500	
		Δp = +1		Δp = −1		Δp = −2		Δp = −3		Δp = −4		Δp = −5		Δp = −6	
		ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh	ΔW's	Δh
20	.1027	−0.5	−0.08	0.5	0.08	1.0	0.17	1.7	0.26	2.2	0.36	3.0	0.46	3.8	0.56
21	.1078	−0.5	−0.08	0.5	0.08	1.1	0.17	1.8	0.27	2.4	0.37	3.2	0.49	4.0	0.61
22	.1130	−0.5	−0.08	0.6	0.09	1.1	0.18	1.9	0.28	2.6	0.40	3.4	0.52	4.2	0.64
23	.1186	−0.6	−0.09	0.6	0.09	1.2	0.19	2.0	0.30	2.7	0.41	3.6	0.55	4.4	0.67
24	.1243	−0.6	−0.09	0.6	0.10	1.3	0.20	2.1	0.32	2.8	0.43	3.7	0.57	4.6	0.71
25	.1303	−0.6	−0.10	0.7	0.10	1.4	0.21	2.2	0.33	3.0	0.46	3.9	0.60	4.9	0.75
26	.1366	−0.7	−0.10	0.7	0.11	1.4	0.22	2.3	0.35	3.1	0.48	4.1	0.63	5.1	0.78
27	.1431	−0.7	−0.11	0.7	0.11	1.5	0.23	2.4	0.37	3.3	0.50	4.3	0.66	5.4	0.82
28	.1500	−0.7	−0.11	0.8	0.12	1.6	0.24	2.5	0.38	3.4	0.52	4.5	0.69	5.6	0.86
29	.1571	−0.8	−0.12	0.8	0.12	1.7	0.26	2.6	0.40	3.6	0.55	4.7	0.72	5.8	0.89
30	.1645	−0.8	−0.12	0.8	0.13	1.7	0.27	2.7	0.42	3.8	0.58	4.9	0.75	6.1	0.93
31	.1723	−0.8	−0.13	0.9	0.13	1.8	0.28	2.8	0.44	3.9	0.60	5.1	0.78	6.3	0.97
32	.1803	−0.9	−0.13	0.9	0.14	1.9	0.29	3.0	0.45	4.1	0.63	5.3	0.81	6.6	1.01
33	.1878	−0.9	−0.14	1.0	0.15	2.0	0.30	3.1	0.47	4.3	0.66	5.5	0.85	6.9	1.06
34	.1955	−0.9	−0.14	1.0	0.15	2.1	0.32	3.2	0.49	4.4	0.68	5.7	0.88	7.2	1.11
35	.2034	−1.0	−0.15	1.0	0.16	2.1	0.33	3.4	0.51	4.6	0.71	6.0	0.92	7.5	1.16
36	.2117	−1.0	−0.15	1.1	0.17	2.2	0.35	3.5	0.53	4.8	0.74	6.2	0.96	7.8	1.20
37	.2202	−1.0	−0.16	1.1	0.17	2.3	0.36	3.6	0.56	5.0	0.77	6.5	1.00	8.1	1.25
38	.2290	−1.1	−0.17	1.2	0.18	2.4	0.37	3.8	0.58	5.2	0.80	6.8	1.05	8.4	1.30
39	.2382	−1.1	−0.18	1.2	0.19	2.5	0.39	3.9	0.61	5.4	0.85	7.1	1.09	8.8	1.36
40	.2477	−1.2	−0.18	1.3	0.20	2.6	0.41	4.1	0.63	5.7	0.88	7.4	1.14	9.2	1.42
41	.2575	−1.2	−0.19	1.3	0.20	2.7	0.42	4.3	0.66	5.9	0.91	7.7	1.19	9.6	1.48
42	.2676	−1.3	−0.20	1.4	0.21	2.8	0.44	4.4	0.69	6.1	0.94	8.0	1.23	10.0	1.54
43	.2781	−1.3	−0.21	1.4	0.22	3.0	0.45	4.6	0.71	6.4	0.99	8.4	1.29	10.4	1.61
44	.2890	−1.4	−0.22	1.5	0.23	3.1	0.47	4.8	0.74	6.7	1.04	8.7	1.34	10.8	1.67
45	.3002	−1.4	−0.22	1.5	0.24	3.2	0.49	5.0	0.77	6.9	1.07	9.1	1.40	11.3	1.75
46	.3119	−1.5	−0.23	1.6	0.25	3.3	0.51	5.2	0.80	7.2	1.11	9.4	1.45	11.7	1.81
47	.3239	−1.6	−0.24	1.7	0.26	3.4	0.53	5.4	0.84	7.5	1.16	9.8	1.52	12.1	1.87
48	.3363	−1.6	−0.25	1.8	0.27	3.6	0.56	5.6	0.87	7.8	1.21	10.2	1.58	12.6	1.95
49	.3491	−1.7	−0.26	1.8	0.28	3.7	0.58	5.8	0.90	8.1	1.25	10.5	1.63	13.1	2.03
50	.3624	−1.7	−0.27	1.9	0.29	3.9	0.60	6.1	0.94	8.4	1.30	10.9	1.69	13.6	2.11
51	.3761	−1.8	−0.28	2.0	0.30	4.0	0.62	6.3	0.97	8.7	1.35	11.3	1.75	14.1	2.18
52	.3903	−1.9	−0.29	2.0	0.31	4.2	0.64	6.5	1.01	9.0	1.40	11.8	1.82	14.7	2.28
53	.4049	−1.9	−0.30	2.1	0.32	4.4	0.68	6.7	1.05	9.3	1.44	12.2	1.89	15.2	2.36
54	.4200	−2.0	−0.31	2.2	0.34	4.5	0.70	7.0	1.09	9.7	1.50	12.7	1.97	15.8	2.45
55	.4356	−2.1	−0.32	2.3	0.35	4.7	0.73	7.3	1.13	10.1	1.57	13.2	2.05	16.4	2.54
56	.4518	−2.2	−0.34	2.4	0.37	4.9	0.76	7.6	1.18	10.5	1.63	13.7	2.13	17.1	2.64
57	.4684	−2.3	−0.35	2.4	0.37	5.1	0.79	7.9	1.22	10.9	1.69	14.2	2.21	17.7	2.75
58	.4856	−2.3	−0.37	2.5	0.39	5.3	0.82	8.2	1.27	11.3	1.76	14.7	2.30	18.4	2.86
59	.5033	−2.4	−0.38	2.6	0.41	5.4	0.85	8.5	1.32	11.7	1.82	15.3	2.38	19.1	2.97
60	.5216	−2.5	−0.40	2.7	0.42	5.7	0.88	8.8	1.37	12.2	1.90	15.9	2.47	19.9	3.09
61	.5405	−2.6	−0.41	2.8	0.44	5.9	0.91	9.2	1.43	12.7	1.98	16.5	2.57	20.7	3.22
62	.5599	−2.7	−0.43	2.9	0.46	6.1	0.95	9.5	1.48	13.2	2.05	17.1	2.66	21.4	3.33
63	.5800	−2.8	−0.44	3.0	0.48	6.3	0.98	9.9	1.54	13.7	2.13	17.7	2.76	22.3	3.47
64	.6007	−2.9	−0.46	3.2	0.49	6.5	1.02	10.2	1.59	14.2	2.21	18.4	2.87	23.1	3.60
65	.6221	−3.1	−0.48	3.3	0.51	6.8	1.06	10.6	1.65	14.7	2.29	19.1	2.98	23.9	3.72
66	.6441	−3.2	−0.50	3.4	0.53	7.1	1.10	11.0	1.72	15.3	2.38	19.8	3.09	24.8	3.87
67	.6668	−3.3	−0.51	3.5	0.55	7.3	1.14	11.4	1.78	15.8	2.47	20.5	3.20	25.7	4.01
68	.6902	−3.4	−0.53	3.7	0.57	7.6	1.18	11.8	1.84	16.4	2.56	21.3	3.32	26.7	4.16
69	.7143	−3.5	−0.55	3.8	0.59	7.9	1.22	12.2	1.90	17.0	2.65	22.1	3.45	27.7	4.32
70	.7392	−3.7	−0.57	3.9	0.61	8.1	1.27	12.7	1.98	17.6	2.75	22.9	3.58	28.7	4.48
71	.7648	−3.8	−0.59	4.1	0.64	8.4	1.32	13.1	2.05	18.2	2.84	23.7	3.70	29.7	4.64
72	.7911	−3.9	−0.61	4.2	0.66	8.7	1.36	13.6	2.13	18.8	2.94	24.6	3.84	30.8	4.82
73	.8183	−4.1	−0.63	4.4	0.69	9.0	1.41	14.1	2.20	19.5	3.05	25.5	3.99	31.9	4.99
74	.8463	−4.2	−0.66	4.6	0.71	9.4	1.46	14.6	2.28	20.2	3.16	26.4	4.14	33.1	5.18
75	.8750	−4.4	−0.68	4.7	0.74	9.7	1.52	15.1	2.36	20.9	3.27	27.4	4.28	34.3	5.37
76	.9047	−4.5	−0.71	4.9	0.77	10.0	1.57	15.7	2.46	21.7	3.39	28.3	4.42	35.5	5.56
77	.9352	−4.7	−0.73	5.1	0.79	10.4	1.63	16.3	2.55	22.5	3.52	29.4	4.61	36.9	5.77
78	.9667	−4.9	−0.76	5.3	0.82	10.8	1.69	16.9	2.65	23.3	3.65	30.5	4.77	38.2	5.98
79	.9990	−5.0	−0.79	5.4	0.85	11.2	1.75	17.5	2.74	24.2	3.79	31.6	4.95	39.6	6.20
80	1.032	−5.2	−0.82	5.6	0.88	11.6	1.82	18.1	2.84	25.1	3.93	32.7	5.13	41.0	6.43
81	1.067	−5.4	−0.85	5.8	0.91	12.0	1.88	18.8	2.95	26.0	4.08	33.9	5.32	42.5	6.66
82	1.102	−5.6	−0.88	6.0	0.94	12.5	1.96	19.5	3.06	27.0	4.24	35.1	5.51	44.0	6.90
83	1.138	−5.8	−0.91	6.2	0.97	12.9	2.03	20.2	3.17	28.0	4.39	36.4	5.71	45.6	7.15
84	1.175	−6.0	−0.94	6.4	1.00	13.3	2.10	20.9	3.28	28.9	4.54	37.7	5.92	47.3	7.41

t = Dry bulb temperature (F).
t' = Wet bulb temperature (F).
p = Barometric pressure (in. of Hg).
Δp = Pressure difference from standard barometer (in. of Hg).
W = Specific humidity of air (gr per lb of dry air).
W'_s = Specific humidity of air saturated at wet bulb temperature t' (gr per lb of dry air).
ΔW = Specific humidity correction of air when barometric pressure differs from standard barometer (gr per lb of dry air).
$\Delta W'_s$ = Specific humidity correction of air saturated at wet bulb temperature when barometric pressure differs from standard barometer (gr per lb of dry air).

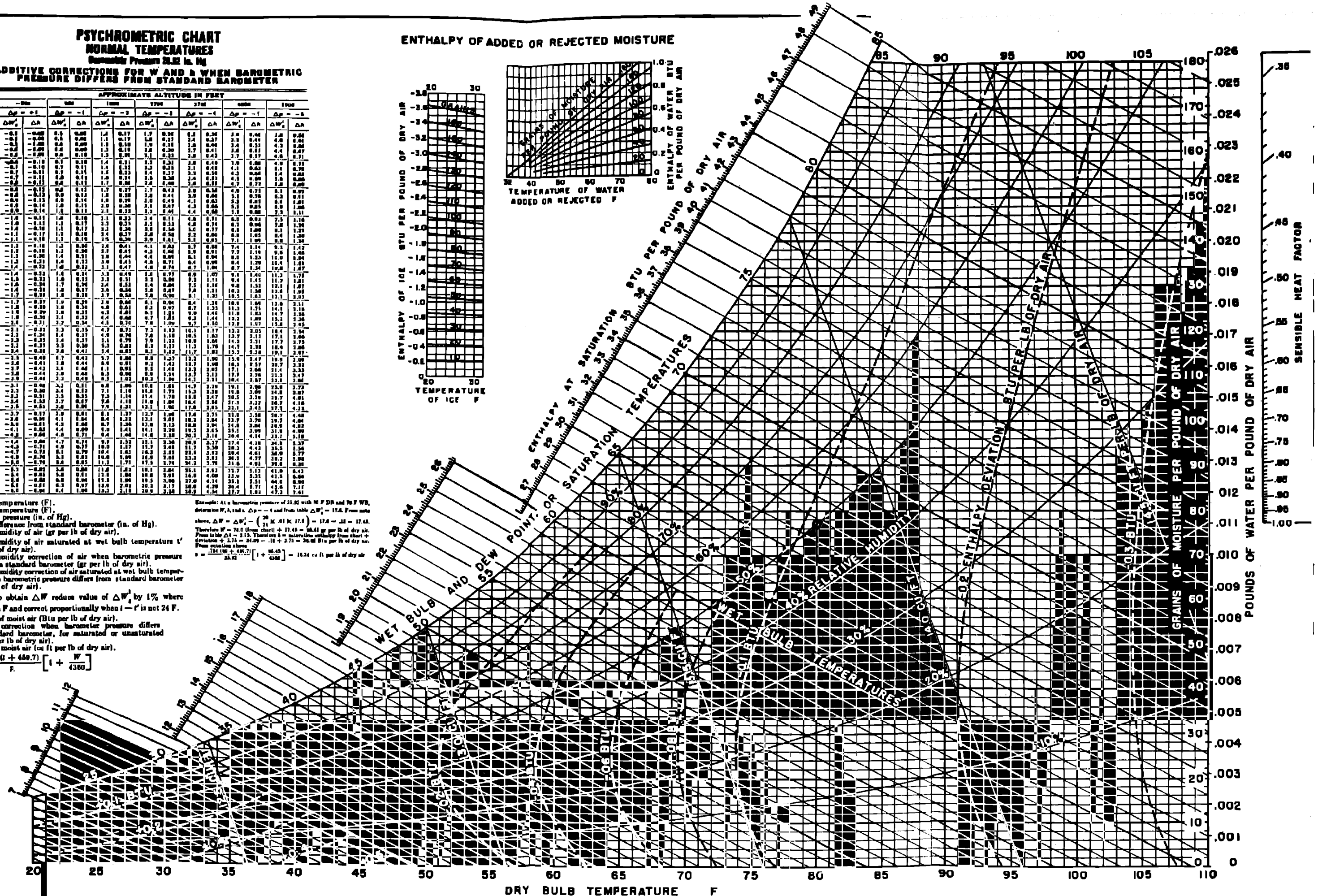
NOTE: To obtain ΔW reduce value of $\Delta W'_s$ by 1% where $t - t' = 24$ F and correct proportionally when $t - t'$ is not 24 F.

h = Enthalpy of moist air (Btu per lb of dry air).
Δh = Enthalpy correction when barometer pressure differs from standard barometer, for saturated or unsaturated air (Btu per lb of dry air).
v = Volume of moist air (cu ft per lb of dry air).

$$v = \frac{.754\,(t + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$

Example: At a barometric pressure of 25.92 with 90 F DB and 70 F WB, determine W, h, and v. Δp = −4 and from table $\Delta W'_s$ = 17.6. From note above, $\Delta W = \Delta W'_s - \left(\frac{20}{24} \times .01 \times 17.6\right) = 17.6 - .15 = 17.45$. Therefore W = 78.0 (from chart) + 17.45 = 95.45 gr per lb of dry air. From table Δh = 2.75. Therefore h = saturation enthalpy from chart + deviation + 2.75 = 34.09 − .10 + 2.75 = 36.74 Btu per lb of dry air. From equation above

$$v = \frac{.754\,(90 + 459.7)}{25.92}\left[1 + \frac{95.45}{4360}\right] = 16.34 \text{ cu ft per lb of dry air}$$



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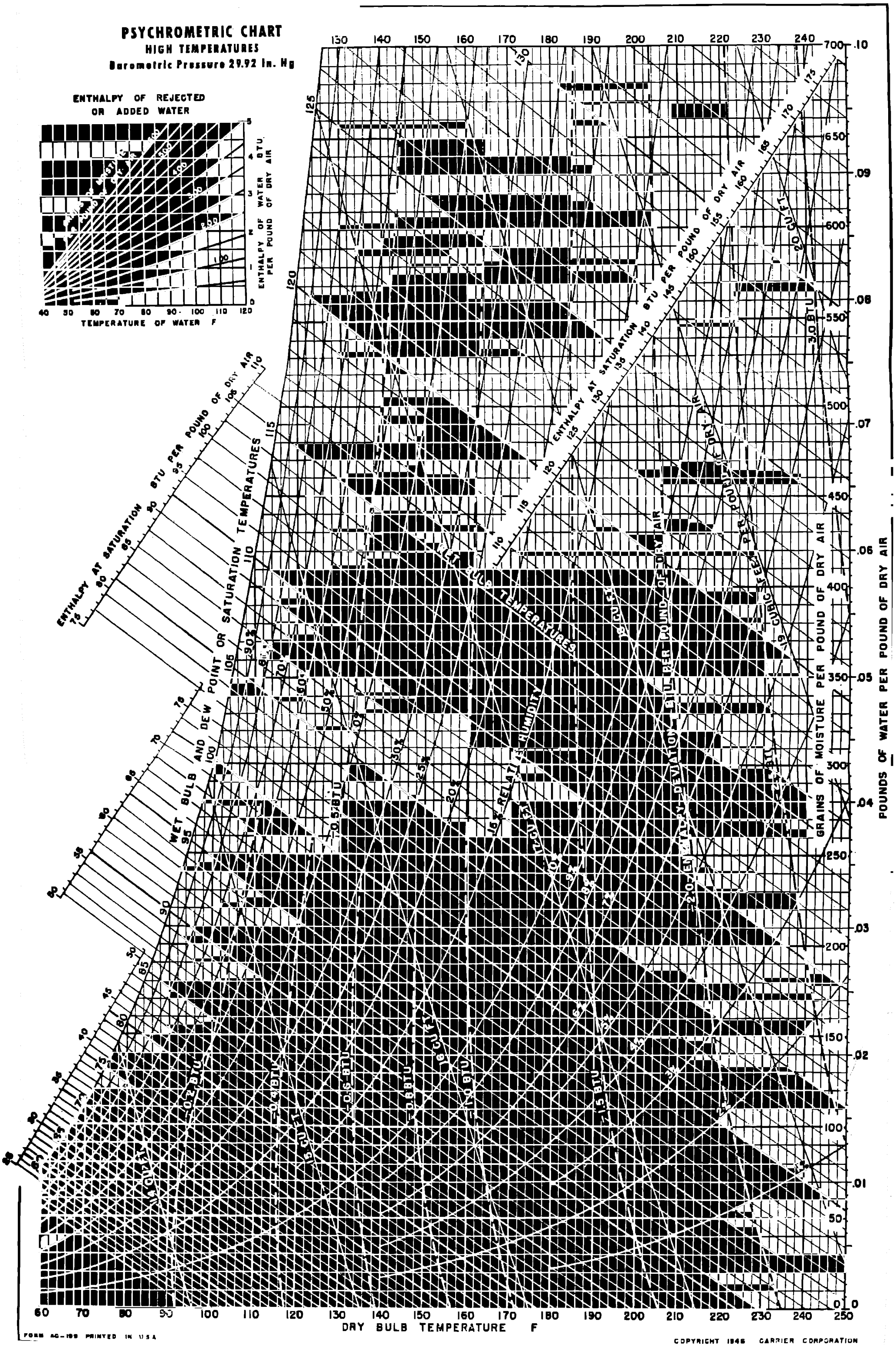
PSYCHROMETRIC CHART

HIGH TEMPERATURES

Barometric Pressure 29.92 In. Hg



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actual vapor pressure (.555 in. Hg) by the vapor pressure corresponding to saturation at 90 F (1.422 in. Hg) to obtain relative humidity of 39.0%.

Volume of one pound of dry air with contained moisture is determined from:

$$v=\frac{.754(t+460)}{p}\left[1+\frac{W}{4360}\right]$$

$$=\frac{.754(90+460)}{25.92}\left[1+\frac{95.5}{4360}\right]$$

$$=16.4 \text{ cu ft per lb of dry air}$$

III. The Measurement of Moist Air Properties[5]

Humidity measurements in comfort air conditioning and industrial processing are made almost exclusively by the **wet bulb psychrometer,** consisting of a combination of wet bulb and dry bulb thermometer placed in relative motion to the air. In meteorology, the psychrometer finds universal use even at the extremely low temperature of Arctic regions. The psychrometer may also be used in engineering practice at temperatures below freezing if suitable precautions are taken in its use.

Of the several methods available for measuring humidity at low temperatures, the psychrometer combines the advantages of simplicity, low cost, ease of manipulation and commercial availability. When properly used, it is entirely reliable.

The purpose here is to (1) recommend the practices and precautions necessary to obtain reliable readings with the wet bulb psychrometer, and (2) establish the magnitude of errors that may exist under various operating conditions.

3.1—Selection of Thermometers

As in any other process, the precision required for thermometers used in psychrometry will depend upon the property being determined and the degree of accuracy required in the results. It so happens that the permissible error in thermometer readings can be easily estimated.

Regardless of temperature range, the wet-bulb reading is generally more important than the dry-bulb reading. In the comfort air conditioning range, for example, an error in the wet bulb reading will indicate an error in specific humidity approximately three times as great as a similar error in the dry bulb reading. This ratio decreases with temperature, until at very low temperatures the dry bulb and wet bulb readings approach equal importance. The determination of enthalpy depends almost entirely upon the wet bulb reading.

For any particular process, the required accuracy of wet bulb reading will depend upon the overall change in wet bulb temperature. In air conditioning practice, it is not uncommon to work with changes in the wet-bulb temperature in the order of 5 to 10 F between air entering and leaving the equipment. Thus for a process involving a change of 5 F in the wet-bulb temperature, it can be assumed that an error in the wet bulb reading of 0.1 F will result in an error of approximately 2% in either specific humidity difference or enthalpy difference. Air coolers operating in the low temperature range generally have a change in **dry bulb temperature** between entering and leaving conditions that is considerably less than would be used in the comfort air conditioning range. However, it will be found that the change in **wet bulb temperature** again runs in the range of 5 F. Thus for the testing of air coolers, a thermometer accuracy of 0.1 F is generally required in order to insure that the error in enthalpy difference or specific humidity difference does not exceed 2%.

For processes covering a wide range between entering the leaving wet bulb temperatures, the higher wet bulb temperature becomes more important than the lower one, as will be noted by referring to **Table 2.** This table, which shows the deviation of enthalpy and specific humidity corresponding to various errors in the wet bulb readings at various temperature ranges, is useful for determining the accuracy of wet-bulb reading necessary to achieve a specified accuracy of enthalpy or specific humidity and in turn for achieving a specified percent of accuracy for the process.

Notes on the wet bulb wicks. A suitable material for the wick is **cotton tubing** of a fairly soft, fine mesh weave. Before use, the wick should be thoroughly cleaned by washing or boiling. A snug fit on the thermometer bulb is necessary and in order to prevent excessive conduction of heat from the stem, the wick should cover about one inch of the stem as well as the bulb. With continued use, wicks become encrusted with impurities that interfere with the proper action. It is, therefore, highly important that wicks be frequently cleaned or replaced.

Table 2

Wet bulb temp, F	Deviation of enthalpy and specific humidity, corresponding to various errors in the wet-bulb reading			
	1.0 F Error	0.5 F Error	0.2 F Error	0.1 F Error
80	1.08 Btu 5.3 gr	.54 Btu 2.6 gr	.22 Btu 1.1 gr	.11 Btu .5 gr
60	.7 Btu 2.8 gr	.35 Btu 1.4 gr	.14 Btu .56 gr	.07 Btu .03 gr
40	.46 Btu 1.4 gr	.23 Btu .70 gr	.09 Btu .28 gr	.05 Btu .14 gr
20	.35 Btu .7 gr	.18 Btu .35 gr	.07 Btu .14 gr	.04 Btu .07 gr
0	.28 Btu .3 gr	.14 Btu .15 gr	.05 Btu .06 gr	.03 Btu .03 gr
−20	.26 Btu .1 gr	.13 Btu .05 gr	.05 Btu .02 gr	.03 Btu .01 gr

The ice-coated bulb. The wick covering for the wet-bulb thermometer at temperatures below freezing no longer serves its usual purpose, for ice, unlike water, does not respond to capillary forces. True wet-bulb readings are obtained only when the surface of the wick is completely covered with a layer of ice. The ice held within the wick is useless.

As a result of the reduced vapor pressure at low temperatures, a longer time is necessary to reach equilibrium than at higher temperatures. This condition is offset, however, by the ice remaining on the bulb for a much longer period of time. Readings of the wet-bulb thermometer must be continued over a sufficiently long period to insure that equilibrium has been reached.

Careful tests indicate that reliable results are obtained by discarding the wick and freezing a layer of ice directly on the thermometer bulb. An ice film 0.02 in. thick has been found to have a life of roughly one hour when exposed to an air velocity of 900 fpm with a saturation deficiency of about 1.4 gr of moisture. The ice film is best formed by dipping the chilled thermometer into distilled water at approximately 32 F. The thermometer is then removed from the water and the film allowed to freeze. The process may be repeated several times if necessary to build up a suitable film thickness.

There is evidence[6] that the mechanical expansion of the ice in freezing might set up stresses in certain types of thermometers to cause an error. One method of overcoming this condition is to build a tube the same size as the thermometer bulb, freeze a coating of ice over it, thaw it with warm water inside the tube, and place the cup of ice over the thermometer.

In order to prevent excessive conduction of heat along the glass stem of the thermometer to the bulb, it is important that the ice film cover about one inch of the stem as well as the bulb.

3.2—Super-cooled Water

The tendency of water to resist freezing at temperatures below freezing is well known. This condition of super-cooled water can and does exist on the wick of the wet-bulb thermometer and may give rise to considerable error. While it is possible by the aid of a special psychrometric chart to obtain correct readings of psychrometric properties with the wick wetted with super-cooled water, it is generally more desirable to coat the bulb with ice.

3.3—Effect of Air Stream Velocity

In practice the wet bulb thermometer does not always read the true thermodynamic wet-bulb temperature and under certain conditions may deviate considerably from it. The relative rates of diffusion of moisture from the wetted surface as compared to the diffusion of the surrounding air to the wetted surface tends to make the actual wet bulb temperature lower than the thermodynamic wet bulb temperature. Heat received by radiation from surrounding objects at dry bulb temperature tends to make the wet bulb thermometer read high. Fortunately, these two effects tend to balance each other, with the result that the wet bulb thermometer may correspond to the thermodynamic wet bulb temperature at certain air velocities, and will generally deviate from the theoretical values less than if either one of these effects were present without the other.

Fig. 10 shows the error in the wet bulb depression for various air velocities. This curve may be used to correct the reading of wet bulb thermometers where it is impractical to maintain a suitably high velocity. It should be noted that for temperatures in the comfort air conditioning range, a velocity of approximately 1,000 fpm is desirable for minimum error. At lower temperatures, a considerably lower velocity is indicated.

Note that the curves shown in Fig. 10 apply to a thermometer bulb of conventional size (slightly under ¼ in. diameter). For other diameters the velocity scale will be directly proportional to the diameter.

Radiation from coils and other objects. If the wet bulb thermometer is located close to cooling or heating surfaces, there may result an error due to radiation from such surfaces. In order to avoid this error, a shield constructed from polished metal should be placed between the thermometer and the radiating surface.

Wet bulb thermometers located in air ducts should not be shielded from the duct wall if the wall is at or near dry bulb temperature. If necessary, the outside of the duct should be insulated to avoid a large difference between the duct wall and the inside dry-bulb temperatures.

Radiation from the observer. If the observer enters the refrigerated space, precaution must be taken to avoid radiation from the observer's body to the wet bulb and dry bulb thermometers. Furthermore, heat from the observer's body and his exhalation of breath may change the temperature and relative humidity in the vicinity in which the readings are being taken. Reliable results can generally be obtained if the observer keeps in motion while using a sling psychrometer and takes readings quickly while avoiding breathing on the thermometers.

3.4—Use of Thermocouples

Thermocouples are entirely satisfactory for taking psychrometric readings if suitable equipment is available and proper precautions are taken in its use. The wet and dry bulb thermocouples may be connected

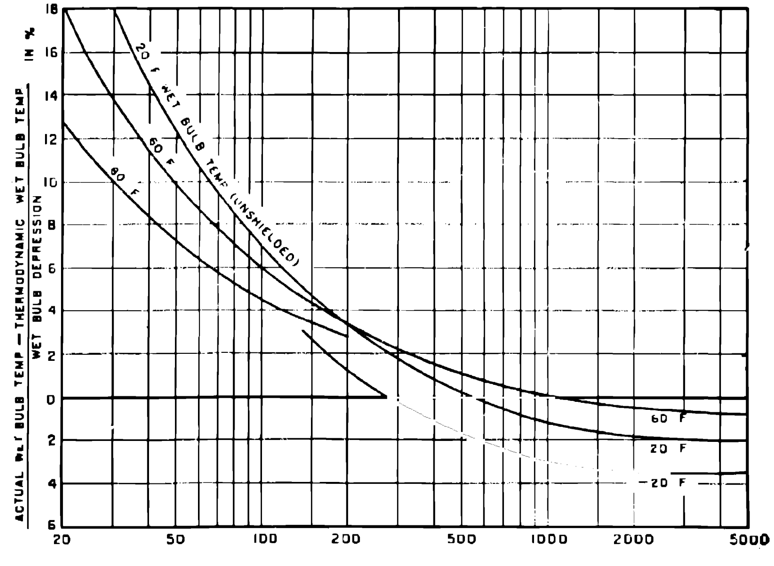

Fig. 10. The Effect of Air Stream Velocity on the Wet-Bulb Psychrometer (For conventional size thermometer bulb)

in series to give a direct reading of the depression and, when measured on a potentiometer of suitable accuracy, such a series couple forms one of the most reliable methods of obtaining psychrometric data.

Thermocouple wires most commonly used in refrigeration practice are copper and constantan or iron and constantan. The author prefers copper-constantan for freedom from rust. Copper has a high thermal conductivity and may introduce errors due to the conduction of heat from the surrounding air, at dry-bulb temperature, to the lower wet-bulb temperature. This condition is best remedied by selecting the copper wire about twelve sizes smaller than the constantan. For example, No. 36 copper wire should be selected for use with No. 24 constantan.

The wick must fit snugly over the thermocouple wire. Small size cotton tubing can be obtained for this purpose. For No. 24 constantan and No. 36 copper wire, the wick should cover about one inch of the lead wires beyond the junction, care being taken to insulate the leads from each other to prevent short circuiting. With larger size wire, a much longer length of lead must be included within the wick. At temperatures below freezing, the wick may be discarded and ice frozen directly on the thermocouples.

The small size of the thermocouple makes it particularly suited for psychrometric determinations at low air stream velocities since as previously stated the velocity scale of Fig. 10 is proportional to diameters. Thus a wet-bulb thermocouple having a diameter of 1/16 in. will have the same correction at 100 fpm as the ¼-in. thermometer has at 400 fpm.

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5. HEAT TRANSMISSION

Introduction

1. Heat always tends to flow from a source of high temperature to a substance at a lower temperature. Heat may be transmitted from one temperature level to a lower temperature level by three different methods—conduction, convection and radiation. **Conduction** denotes the flow of heat from one part of a body to another part of the same body or to another body in contact, without displacement of the particles. In conduction, heat is transmitted by molecular motion. The molecules in the material where heat is applied are made to vibrate more rapidly as they are heated, and they transmit their energy as energy of motion to adjoining molecules which in turn are sped up and affect their neighbors, raising their temperatures without any appreciable displacement of material.

Heat transmission through fluids is called **convection.** Actually, there is conduction present when heat is transmitted through fluids, but when the fluid is not prevented from flowing, the transmission is said to be caused by convection. Usually, "free" or "natural" convection is caused by a physical change in the material being heated or cooled. Thus, when water is heated in a pot, the water is first warmed at the bottom and an expansion takes place, making the heated water less dense than the cooler water at the top. The warm water flows upward and the cooler water flows downward, causing "convection currents." This mixing action serves to carry the heat to all portions of the fluid. If the fluid is restricted or the heat is applied in such a manner that free flow does not result, then heat is transmitted only by conduction. If a test tube of water is held over a flame so that only the upper part is heated, there are no forces set up to cause the fluid to flow, so the bottom of the test tube will stay cool until actual conduction has heated it. When the fluid motion is caused by an external force, a fan or pump, for example, the heat interchange is called "forced" convection.

In a sense convection really permits conduction by removing the heated fluid, allowing heat to flow to an unheated portion of the fluid. At the boundary layer at the edge of the fluid, frictional and viscous forces will tend to hold a relatively small layer of the fluid at rest or at a very low velocity. This boundary layer is known as a "fluid film," and heat is transmitted through this film to the main body of the fluid essentially by conduction.

A special case of heat transmission by convection occurs when evaporation or condensation takes place. In a closed container, where vapor and liquid exist in equilibrium, all parts tend to remain at a constant temperature, for if a given part becomes cooler, its local vapor pressure will be reduced. Surrounding vapor pressure will then force more vapor to the cool spot and the vapor will condense, releasing its latent heat and tending to warm the spot. Refrigerant condensers and pressure-cookers provide examples of this form of transmission.

The transmission of heat by **radiation** involves the direct transfer of energy from one body to another through space, at any distance and without the need for intervening matter. It is an electromagnetic wave phenomenon similar to light, but of longer wave length.

All bodies give off heat in the form of radiant energy, and when a system of bodies at different temperatures exists, each body will radiate and absorb heat, in amounts depending on the composition of the bodies and their surface conditions and temperatures, until eventually an equilibrium will exist, when all bodies are at the same temperature. When equilibrium has been attained, the radiation continues, but then the radiation and absorption are equal, so there is no further temperature change.

Nomenclature

2. The following **nomenclature** will be used throughout this chapter:

A = area perpendicular to the direction of heat flow, sq ft
c = specific heat of fluid, Btu/(lb) (deg F); c_p at constant pressure
C = conductance, Btu/(hr) (deg F)
D = diameter, ft; D_e equivalent diameter
d = prefix indicating differential; dimensionless
e = base of natural logarithm = 2.718.
G = mass velocity, lb/(hr) (sq ft of cross section)
g = acceleration due to gravity, 4.17×10^8 ft/(hr) (hr)
h = heat transfer coefficient, Btu/(hr) (sq ft) (deg F)
k = thermal conductivity, Btu/(hr) (sq ft) (deg F per ft)
L = length, ft
$\log_e$ = natural logarithm = 2.303 $\log_{10}$
Q = quantity of heat, Btu
q = rate of heat flow, Btu/hr; q_c by convection and conduction, q_r by radiation
R = thermal resistance (deg F) (hr)/Btu
t = temperature, deg F
T = absolute temperature, deg F abs = $460+t$
U = overall coefficient of heat transfer Btu/(hr) (sq ft) (deg F)
V = velocity, ft per hr = G/ρ
x = length of conduction path, ft

Greek

β (Beta) = coefficient of volumetric expansion, for gases = 1/deg F abs
Δ (Delta) = symbol indicating difference; dimensionless
ϵ (Epsilon) = emissivity; dimensionless
θ (Theta) = time, hr
λ (Lambda) = latent heat of condensation, Btu/lb
μ (Mu) = absolute viscosity, lb/(hr) (ft) = centipoises × 2.42
π (Pi) = 3.1416
ρ (Rho) = density, lb per cu ft
ϕ (Phi) = prefix, designating function; dimensionless
Σ (Sigma) = prefix indicating a summation

Conduction

3. The basic equation for the conduction of heat states that the instantaneous rate of heat flow equals the product of the area perpendicular to the direction of flow, the temperature gradient, and a constant, k, known as the **thermal conductivity**:

$$\frac{dQ}{d\theta} = -kdA\frac{dt}{dx} \qquad (1)$$

where dQ = the amount of heat flowing in time $d\theta$

$\frac{dQ}{d\theta}$ = the instantaneous **rate** of heat flow
dA = area perpendicular to the direction of flow
$\frac{dt}{dx}$ = the temperature gradient or change in temperature, t, per unit length, x

For steady state conditions, where the temperature does not vary with time, and for a body of uniform cross-sectional area, the steady rate of heat flow, q, is expressed

$$q = \frac{kA\Delta t}{x} \qquad (2)$$

where x = length of path of heat flow, A is the area through which the heat flows, and k is in consistent units.

When heat is flowing radially through the walls of a cylinder, such as the heat flow through pipe insulation, Equation (2) may still be used, if A_m, the logarithmic mean area, is used for A. A_m may be found from

$$A_m = \frac{A_2 - A_1}{\log_e (A_2/A_1)} = \frac{A_2 - A_1}{2.303 \log_{10} (A_2/A_1)} \qquad (3)$$

When the outside area is not more than twice the inner area ($A_2/A_1 = 2$ or less), the arithmetic mean, $(A_2+A_1)/2$, will be within 4% of the logarithmic mean and may be used without serious error in most engineering work.

The units used for the thermal conductivity, k, frequently cause confusion. Strictly, in the English system k is expressed as Btu per hr per sq ft per unit temperature gradient, deg F per ft of length of path, but in the refrigeration, air conditioning and building professions the value is frequently expressed **per inch** of length of path. However, for consistency, in this chapter we will use

$$k = \text{Btu/(hr) (sq ft) (deg F per ft)} \qquad (4)$$

Care must be used when selecting values of

k from a table to note the units used, since some tables are based on a foot of thickness and others are based upon an inch.

The values of k vary greatly between different materials, being greatest in metals and lowest in gases and insulators, with the values in liquids falling between. Thus, k for copper is about 220, for water about 0.35, for air about 0.015, and for cork about 0.025. Further, k will vary in a given material, depending upon the apparent density; thus in asbestos with a density of 3.65 lb per board ft, $k = 0.135$, and with a density of 2.44, $k = 0.088$. Thermal conductivity also varies with temperature, so mean values should be used in heat transfer problems. For values of k, see Chap. 12, page 229.

Frequently the flow of heat through several bodies in contact with each other is of importance. In this discussion it is convenient to use the term **resistance**, $R = x/kA$. Then Equation (2) may be written

$$q = \frac{\Delta t}{x/kA} = \frac{\Delta t}{R} \tag{5}$$

For steady state flow through a system of bodies in contact, with no losses due to edge effects, it is obvious that the same amount of heat must pass through each body in a given time. Thus

$$q = \frac{\Delta t_1}{R_1} = \frac{\Delta t_2}{R_2} = \frac{\Delta t_3}{R_3} = \cdots \frac{\Delta t_n}{R_n} \tag{6}$$

where R_1, R_2, etc., are the individual resistances of each body, and $\Delta t_1, \Delta t_2$, etc., are their respective temperature differences. From Equation (6), adding $(qR_1 = \Delta t_1)$ $(qR_2 = \Delta t_2)$, etc.

$$q = \frac{\Delta t_1 + \Delta t_2 + \Delta t_3 + \cdots \Delta t_n}{R_1 + R_2 + R_3 + \cdots R_n} = \frac{\Sigma(\Delta t)}{R_T} \tag{7}$$

in which R_T is the total series resistance, the sum of all the individual resistances. Then

$$q = \frac{\Sigma(\Delta t)}{\dfrac{x_1}{k_1A_1} + \dfrac{x_2}{k_2A_2} + \dfrac{x_3}{k_3A_3} + \cdots \dfrac{x_n}{k_nA_n}} \tag{8}$$

In Equation (8), k_1, k_2, etc., should be taken as the mean thermal conductivities, and A_1, A_2, etc., as mean areas perpendicular to the direction of flow. Thus in cylinders, the logarithmic mean areas (Equation 3) should be used for each individual material.

Conductance is a term used to indicate the overall transfer of heat, when conduction, convection and radiation all may be present. It is defined by the equation

$$C = \Sigma q/\Delta t \tag{9}$$

where C = conductance, Σq is the total heat transmission rate, and Δt the total temperature drop. Conductance of a unit area, or unit conductance, C' equals C/A,

$$C' = \Sigma q/A\Delta t \tag{10}$$

When conduction alone is the method of heat transmission, $q = kA\Delta t/x$ (Equation 2), conductance becomes kA/x, the reciprocal of resistance.

When conditions are not steady, as in intermittent heating and cooling, the conduction problem becomes complex. Since such factors as shape, volume, specific heat, density, and thermal conductivity are involved, generalization is difficult. For a numerical method of solution of these problems see Dusinberre, "Numerical Methods for Transient Heat Flow," Trans. ASME, November, 1945, pp. 703–712.

Convection

4. When heat is applied to a fluid, the heat may be transferred not only by conduction, but also by the mixing of the heated portion with the rest of the fluid, which process is known broadly as **convection**. There are two distinct types of convection—**free** or **natural** convection, which is the transfer of heat from a surface to a fluid causing the fluid to flow only because of its change in density, and **forced** convection, when the circulation of the fluid is caused by mechanical means, such as a fan or a pump.

In either case, whenever there is a fluid in contact with a surface, there is a **fluid film**, which is stationary at the surface and moves only at low velocity near the surface. The study of convection is largely a study of the transfer of heat through this film.

It is assumed that all the resistance to heat transfer in a fluid lies in the film, and that once the heat passes through the film turbulence will distribute it throughout the rest of the fluid. It is conduction through a film of indeterminate thickness that produces convection. So, in Equation (2), if the term k/x be replaced by h, the **film coefficient,** we have

$$q = hA\Delta t \qquad (11)$$

Free convection. A general equation for film coefficients for free convection, first proposed by Lorenz in 1881, is reasonably accurate for vertical planes:

$$h_c = \frac{0.55k}{L}\left[\left(\frac{c_p\mu}{k}\right)\left(\frac{L^3\rho^2\beta g\Delta t}{\mu^2}\right)\right]^{1/4} \qquad (12)$$

where symbols are as noted under nomenclature (page 76) and β, the coefficient of expansion, is $1/T$ for ideal gases. For horizontal cylinders, the outside diameter D is used in place of L.

For surfaces where L is greater than about 2 ft, the effect of L is not appreciable, and

$$h_c = 0.13k\left[\left(\frac{c_p\mu}{k}\right)(\rho^2\beta g\Delta t)\right]^{1/4} \qquad (13)$$

For free convection in air, McAdams[5] recommends the following simplified equations:

Horizontal plates, facing upward:

$$h_c = 0.38\ (\Delta t)^{0.25} \qquad (14)$$

Horizontal plates, facing downward:

$$h_c = 0.2\ (\Delta t)^{0.25} \qquad (15)$$

Vertical plates more than 1 ft high:

$$h_c = 0.27\ (\Delta t)^{0.25} \qquad (16)$$

Vertical plates less than 1 ft high:

$$h_c = 0.28\ (\Delta t/L)^{0.25} \qquad (17)$$

Vertical pipes more than 1 ft high, and also horizontal pipes:

$$h_c = 0.27\ (\Delta t/D_0)^{0.25} \qquad (18)$$

In the above, h_c is the film coefficient for convection. Radiation may account for a substantial amount of the total heat transferred, so the total heat would be

$$q = h_cA(t_s - t_a) + h_rA(t_s - t_w) \qquad (19)$$

where h_r is the coefficient of heat transfer by radiation (see Equation (49), page 88), and t_s, t_a, and t_w are the temperatures of the surface in question, the air, and the surrounding walls, respectively.

Forced convection. A general equation for the film coefficient for velocities greater than the critical was developed by Nusselt, by dimensional analysis:

$$\frac{hD}{k} = \phi\left(\frac{DG}{\mu}\right)\psi\left(\frac{c_p\mu}{k}\right) \qquad (20)$$

in which the dimensionless quantities DG/μ and $c_p\mu/k$ are known as Reynolds number and Prandtl's number, respectively. The functions ϕ and ψ are determined experimentally, and G is the mass velocity, pounds of fluid per hour per square foot of cross section. Although there is no abrupt transition from free to forced convection, free convection usually occurs with Reynolds numbers below 2,100, and forced convection usually produces Reynolds numbers over 3,000.

For heating fluids, whose viscosities are not more than twice that of water, with Reynolds numbers above 2,100, McAdams[4] gives

$$\frac{hD}{k} = .023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.4} \qquad (21)$$

which can also be used for cooling, with 0.3 instead of 0.4 as the exponent for $c_p\mu/k$. For high viscosity liquids and Reynolds numbers over 10,000, Sieder & Tate[10] give

$$\frac{hD}{k} = 0.027\left(\frac{\mu}{\mu_s}\right)^{0.14}\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{1/3} \qquad (22)$$

where μ_s is the viscosity at the surface temperature and μ is the viscosity at the bulk temperature.

For diatomic gases, including air, for which $c_p\mu/k = 0.74$, Equation (21) becomes

$$hD/k = 0.02\ (DG/\mu)^{0.8} \qquad (23)$$

For heating or cooling viscous liquids with velocities below the critical (Reynolds number less than 2,100), in horizontal or vertical tubes, the recommended equation is

$$\frac{hD}{k}\left(\frac{\mu_s}{\mu}\right)^{0.14} = 1.86\left[\left(\frac{DG}{\mu}\right)\left(\frac{c_p\mu}{k}\right)\left(\frac{D}{L}\right)\right]^{1/3} \qquad (24)$$

where μ is the viscosity at the bulk temperature, and μ_s the viscosity corresponding to the average inside surface temperature.

For oils in turbulent flow an approximate equation for heating is

$$h = 70V/Z^{0.83} \tag{25}$$

where V is in fps and Z is in centipoises. For cooling oils, reduce h from Equation (25) by 25%.

For gases heated or cooled outside tube bundles, flowing parallel to the tubes, Equation (21) may be used by substituting for D an equivalent diameter equal to four times the hydraulic radius. The hydraulic radius is obtained by dividing the free area between the tubes by the sum of the perimeters of all the tubes.

For annular spaces, use Equation (21), substituting for D an equivalent diameter equal to the difference between the outer and inner diameters of the annular space.

For gases flowing at right angles to single cylinders, McAdams[5] recommends, on the basis of data for both wires and pipes, for any gas, for D_0G/μ_f from 0.1 to 1,000:

$$\frac{hD_0/k_f}{(c_p\mu_f/k_f)^{0.3}} = 0.35 + 0.47\left(\frac{D_0G}{\mu_f}\right)^{0.52} \tag{26}$$

and when $D_0\,G/\mu_f$ has values from 1,000 to 50,000

$$\frac{hD_0/k_f}{(c_p\mu_f/k_f)^{0.3}} = 0.26\left(\frac{D_0G}{\mu_f}\right)^{0.6} \tag{27}$$

which, for air and other diatomic gases, with Prandtl number $= 0.74$, becomes

$$hD_0/k_f = 0.24\ (D_0G/\mu_f)^{0.6} \tag{28}$$

where k_f and μ_f are at film temperature, which is usually taken as a mean between surface and bulk temperatures, and D_0 is the outside diameter.

The same reference recommends

$$\frac{hD_0}{k_f} = 0.33\left(\frac{c_p\mu_f}{k_f}\right)^{1/3}\left(\frac{D_0G_{\max}}{\mu_f}\right)^{0.6} \tag{29}$$

for both liquids and gases flowing normally to banks of staggered tubes, for $D_0G_{\max}/\mu$ over 2,000, where $G_{\max}$ is the mass velocity through the minimum free area between the tubes. For banks in line (not staggered), Equation (29) should be used, with the constant 0.33 reduced to 0.26.

Figs. 1 through **4** give values for film coefficients for water and Freon-12, both inside and outside, single tubes.

Condensing vapors. When a single vapor condenses, the condensate normally wets the tube, and film type condensation occurs. For condensation outside **horizontal** tubes,[6]

$$h = 0.73\left(\frac{k_f^{3}\rho_f^{2}\lambda g}{D\mu_f N\Delta t_m}\right)^{1/4} \tag{30}$$

where Δt_m is the mean temperature difference between the surface temperature and saturation temperature, λ the latent heat of condensation, and N is the number of rows in a vertical plane.

For **vertical** tubes

$$h = 1.13\left(\frac{k_f^{3}\rho_f^{2}\lambda g}{L\mu_f\Delta t}\right)^{1/4} \tag{31}$$

The film coefficient for Freon-12 is roughly 200 to 400 in horizontal tubes, and that for ammonia about 1,000. Clean steam will give coefficients ranging from 1,000 to 3,000 for film type condensation, and when dropwise condensation of steam occurs on highly polished or treated surfaces, coefficients from 7,000 to 14,000 have been obtained.

Boiling liquids. When a liquid is boiled, as the temperature difference between the boiling liquid and the surface is increased the rate of boiling (and the amount of heat flowing) increases, but reaches a maximum with a **critical temperature difference,** above which the rate of boiling decreases. As the temperature difference becomes greater than this critical difference, the vapor formed by boiling acts as an insulator, impeding the transfer of heat. The critical difference for water is about 45 F.[9]

In general, the rate of boiling is increased as the surface becomes rougher, and as the liquid being boiled is agitated. It is increased with an increase of temperature difference up to the critical point, and is reduced by scale and dirt deposits.

For boiling, it is usually necessary to measure overall transfer coefficients for the particular liquid and physical conditions in question, as there are few published data available.

Overall Transmission

5. In most cases in refrigeration and air conditioning, heat is transmitted from one fluid through a solid wall to another fluid. It is known that in such transmission, the rate of heat transfer, dq, is directly proportional to the area perpendicular to the di-

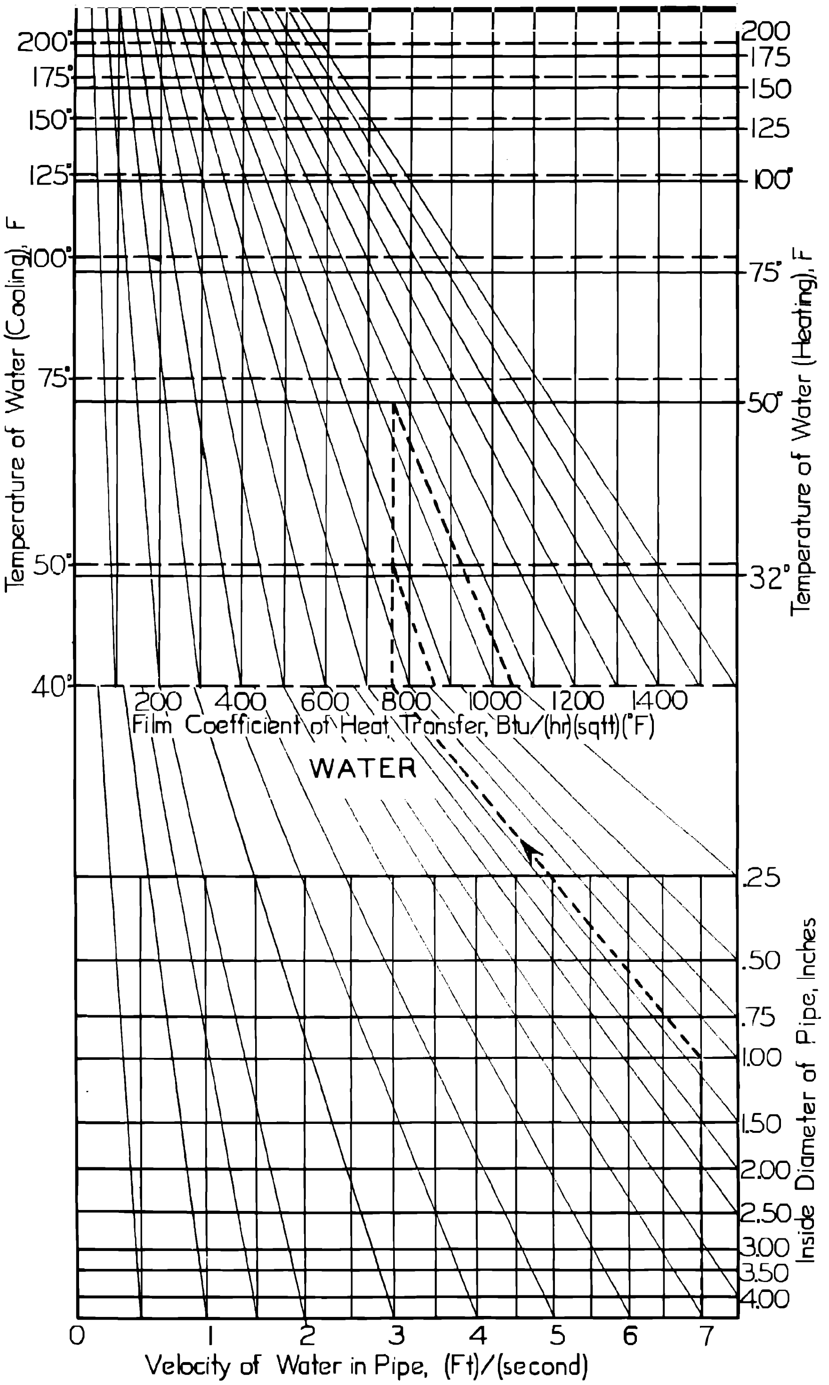

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Fig. 1. Film Coefficient for Water Flowing Inside a Pipe

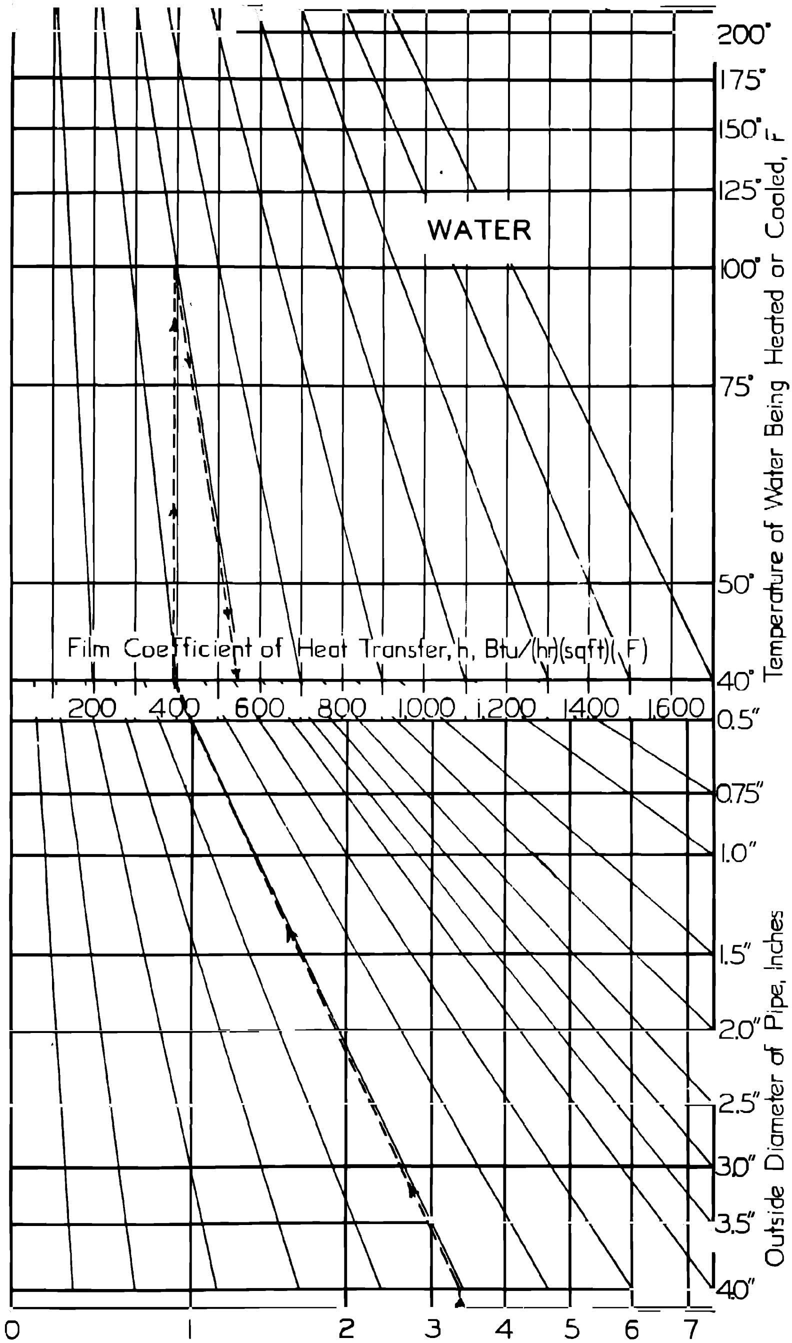

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Fig. 2. Film Coefficient for Water Flowing Outside and Normal to a Single Pipe

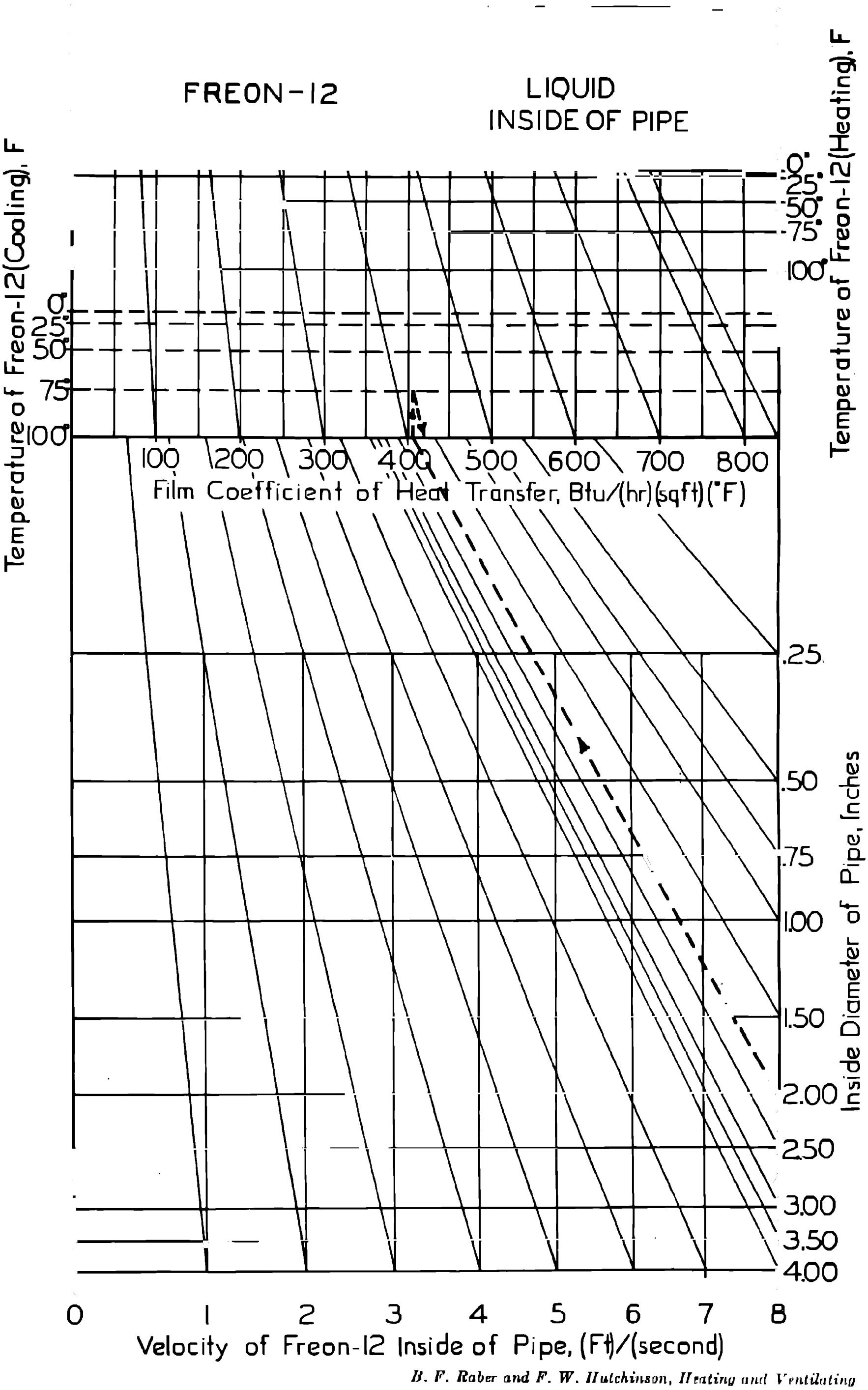

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Fig. 3. Film Coefficient for Liquid Freon-12 Flowing Inside a Pipe

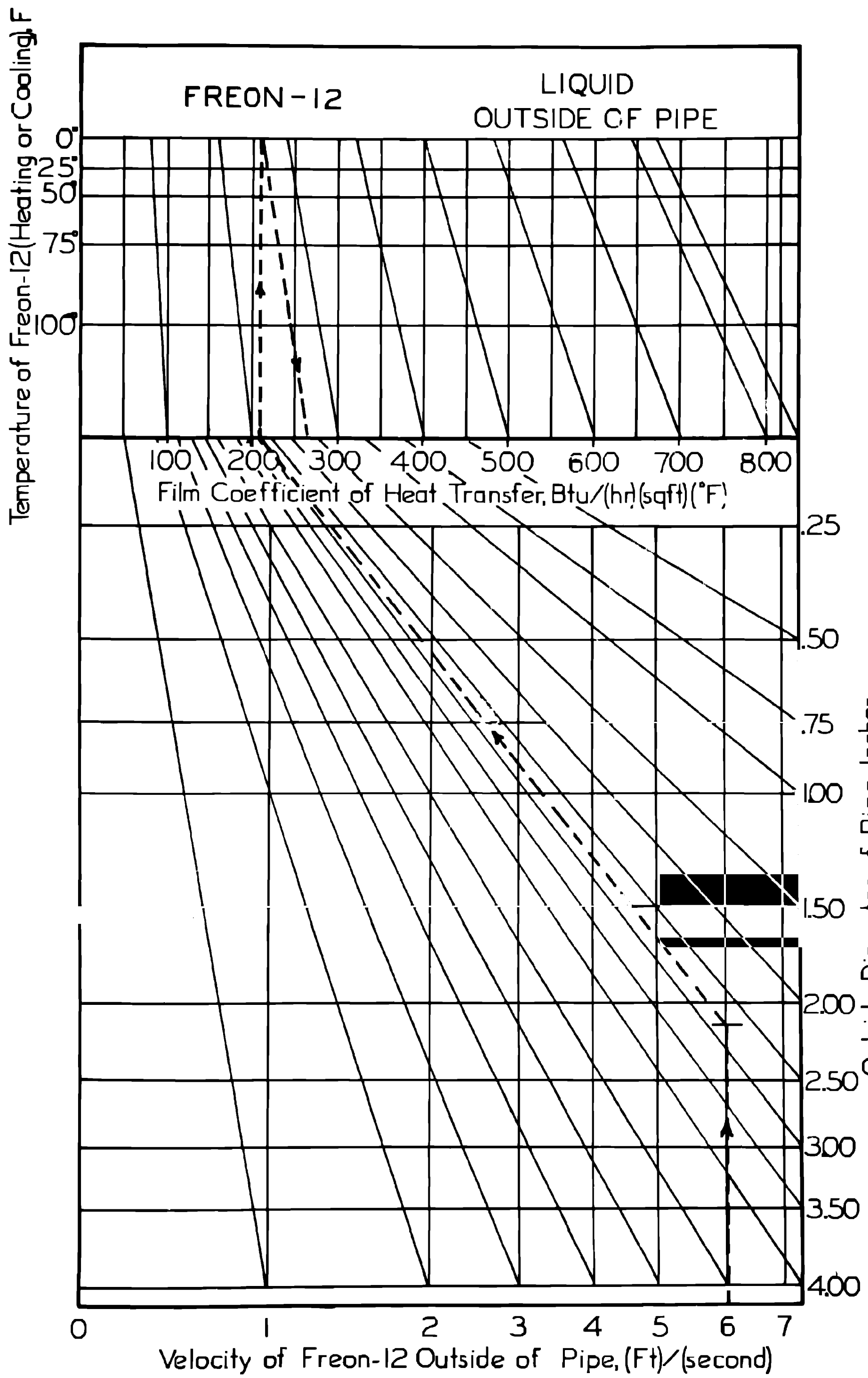

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Fig. 4. Film Coefficient for Liquid Freon-12 Flowing Outside of a Pipe

rection of heat flow, dA, and to the temperature difference existing between the fluids, Δt, or

$$dq = U\,dA\,\Delta t \qquad (32)$$

where U, the proportionality factor, is the **overall coefficient of heat transfer.**

It is obvious that the value of U will depend upon where the measurement of area, dA, is taken (i.e., in a double-pipe interchanger, whether the inside or outside surface of the dividing wall is used.) Any surface can be used provided that the one selected is defined and used consistently.

Equation (32) represents the overall heat transfer; the flow through any single element of the path is given by an equation of the form

$$dq = h'\,dA'\,\Delta t' \qquad (33)$$

in which dA' and $\Delta t'$ are the area of and temperature drop through the element being considered, and h' is called the **local individual coefficient of heat transfer.**

In the case of a double-tube interchanger, with fluids flowing through the center tube and in the annular space between the tubes, if one fluid is warmer than the other there will be a transfer of heat from the warm fluid through the wall to the cooler fluid. Then, from Equation (2) for a wall thickness x_w with a logarithmic mean area dA_w, the rate of conduction through the wall is

$$dq = \frac{k_w dA_w \Delta t_w}{x_w} \qquad (34)$$

Frequently there will be scale or dirt adhering to one of the surfaces of the wall. The same amount of heat passing through the wall, dq, must then also pass through the scale deposit, which has a thickness of x_s, a thermal conductivity k_s, and which will be assumed to be on the inside wall, whose area is dA_i. Since scale deposits are usually thin, the area of the wall is used rather than the log mean area of the scale. So, for the scale,

$$dq = \frac{k_s dA_i \Delta t_s}{x_s} = h_s dA_i \Delta t_s \qquad (35)$$

in which h_s is used for k_s/x_s. (Heat transfer coefficients for scale deposits may be obtained from **Table 1.**[11])

In addition to Equations (34) and (35), we may write equations for the boundary films between the outer fluid and the wall (local coefficient h_0, temperature drop Δt_0), and between the scale and the inner fluid (local coefficient h_i, temperature difference Δt_i).

We have then, for the individual temperature drops:

$$\Delta t_0 = dq/h_0 dA_0$$
$$\Delta t_w = dq\; x_w/k_w dA_w$$
$$\Delta t_s = dq/h_s dA_i$$
$$\Delta t_i = dq/h_i dA_i$$

Adding these four equations gives us the overall temperature difference, Δt:

$$\Delta t = dq\left(\frac{1}{h_0 dA_0} + \frac{x_w}{k_w dA_w} + \frac{1}{h_s dA_i} + \frac{1}{h_i dA_i}\right) \qquad (36)$$

From Equation (32), $\Delta t/dq = 1/U\;dA$, so Equation (36) may be used to find U where dA is the area for which U is desired:

$$\frac{1}{U} = \frac{dA}{h_0 dA_0} + \frac{x_w dA}{k_w dA_w} + \frac{dA}{h_s dA_i} + \frac{dA}{h_i dA_i} \qquad (37)$$

The most common use of this equation is the case where the wall thickness is small compared to the tube diameter, when all areas (dA_i, dA_0, etc.) are very nearly equal. Then

$$\frac{1}{U} = \frac{1}{h_0} + \frac{x_w}{k_w} + \frac{1}{h_s} + \frac{1}{h_i} \qquad (38)$$

If there were scale deposits both inside and outside, and if the tube were made of two materials bonded together, this would become

$$\frac{1}{U} = \frac{1}{h_0} + \frac{1}{h_{s0}} + \frac{x_{w1}}{h_{w1}} + \frac{x_{w2}}{k_{w2}} + \frac{1}{h_{si}} + \frac{1}{h_i} \qquad (39)$$

and since overall resistance R_T for an area dA is $R_T = 1/U$, and correspondingly, $R_0 = dA/h_0 dA_0$, etc., Equation (38) becomes

$$R_T = R_0 + R_w + R_s + R_i \qquad (40)$$

which shows that when heat flows through a series of different resistances, the total resistance is the sum of all individual resistances.

Extended surfaces. Refrigerating engineers have made increasing use of all types of extended surfaces. A thorough analysis of various types of surfaces has been made by Gardner,[2] who produced curves shown in **Figs. 5** through **8.** Values

Table 1. Heat-Transfer Coefficients h_s for Scale Deposits from Water[1]

Temperature of heating medium	Up to 240 F		240–400 F	
Temperature of water	**125 F or less**		**Above 125 F**	
Water velocity, fps	**3 and less**	**Over 3**	**3 and less**	**Over 3**
Distilled	2,000	2,000	2,000	2,000
Sea water	2,000	2,000	1,000	1,000
Treated boiler feed water	1,000	2,000	500	1,000
Treated make-up for cooling tower	1,000	1,000	500	500
City, well, Great Lakes	1,000	1,000	500	500
Brackish, clean river water	500	1,000	330	500
River water, muddy, silty[2]	330	500	250	330
Hard (more than 15 g per gal)	330	330	200	200
Chicago Sanitary Canal	130	170	100	130

[1] From "Standards of Tubular Exchanger Manufacturers Association," 366 Madison Ave., New York, N. Y , 1941.
[2] Delaware, East River (New York), Mississippi, Schuylkill, and New York Bay.

of fin efficiency, ϕ, are given for the types of fins shown on each of the curves where w is the height of the fin in feet, as shown, h is the outside film co-efficient, k the thermal conductivity of the fin material and y_b is half the fin width in feet as shown. However, it is necessary to convert fin efficiency into the actual total resistance of the fin and tube, R_m. Carrier and Anderson[1] give the following formula for obtaining the metal thermal resistance, R_m, in Btu/(hr) (sq ft of total outside surface) (deg F).

$$R_m = \frac{1}{h_a}\left[\frac{1-\phi}{\phi + \dfrac{S_p}{S_f}}\right] \tag{41}$$

where h_a is the outside air film coefficient, S_p is the outside exposed tube surface, sq ft per lineal ft of tube, and S_f is the fin surface, sq ft per lineal ft of tube. Carrier and Anderson[1] also point out that a reasonably close approximation of the efficiency of a square or rectangular plate fin is obtained by assuming a flat circular fin of equal area.

It is necessary, in using the familiar equation

$$q = UA\Delta t_m \tag{42}$$

to understand its limitations. It assumes steady state heat flow, constant specific heats, and adiabatic operation in parallel or counter-flow conditions, with U remaining constant. The mean temperature difference, Δt_m, is properly the logarithmic mean temperature difference (abbreviated log MTD, MED, LMTD, etc.) which is

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \dfrac{\Delta t_1}{\Delta t_2}} = \frac{\Delta t_1 - \Delta t_2}{2.303 \ \log_{10} \dfrac{\Delta t_1}{\Delta t_2}} \tag{43}$$

When U varies with temperature we have the equation by Nikuradse,[7]

$$q = A\left[\frac{U_2\Delta t_1 - U_1\Delta t_2}{\log_e \dfrac{U_2\Delta t_1}{U_1\Delta t_2}}\right] \tag{44}$$

which, when $U_1 = U_2$, becomes equal to Equation (42).

Curves for determining mean temperature differences for reversed current and cross-flow exchangers of different types are given by Bowman, Mueller and Nagle, in Trans. ASME, vol. 62, pp. 283–294, 1940.

Radiation

6. The transmission of heat by radiation is a wave phenomenon, as is the transmission of light; the essential difference between the two is that the wave lengths involved in the transmission of heat are longer than those in light transmission. As the temperature of a body increases the proportionate amount of energy emitted as light increases, but even an incandescent lamp filament radiates only a few per cent of its total energy as light, the greatest portion of the energy being radiated as heat.

Materials differ in their ability to absorb and emit radiation, so a "black body" is used as a reference standard. Defined

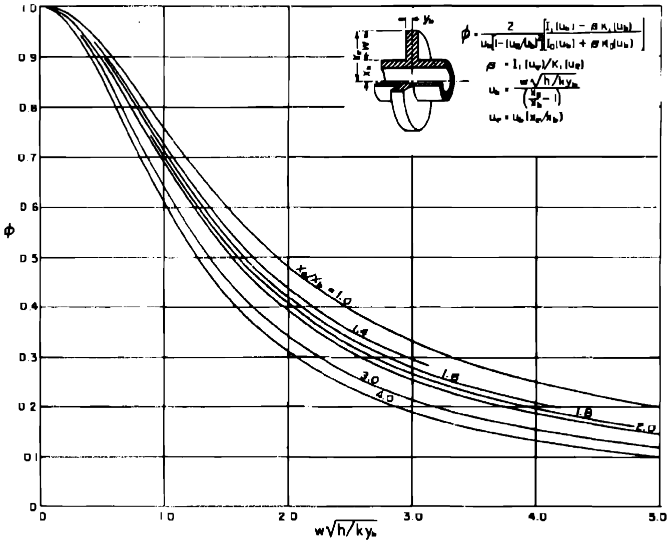

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Fig. 5. Efficiency of Annular Fins of Constant Thickness

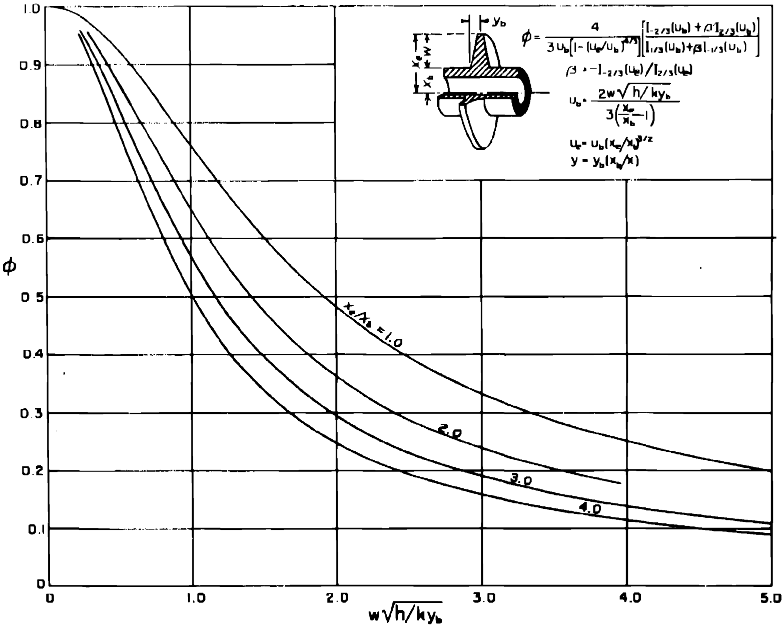

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Fig. 6. Efficiency of Annular Fins with Constant Metal Area for Heat Flow

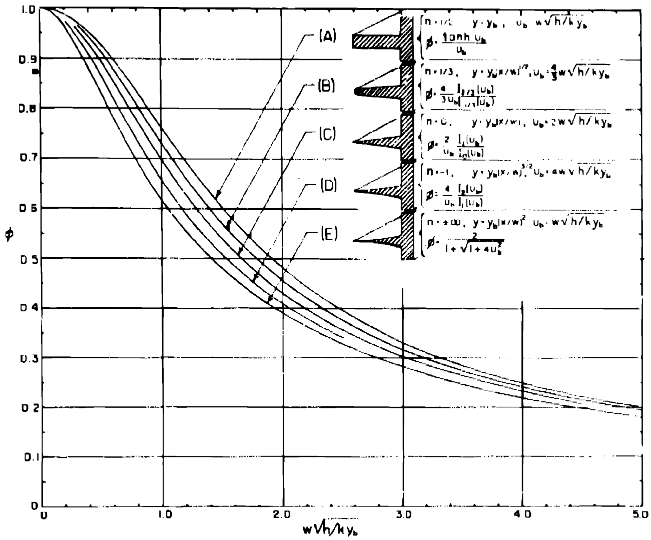

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Fig. 7. Fin Efficiency of Several Types of Straight Fin

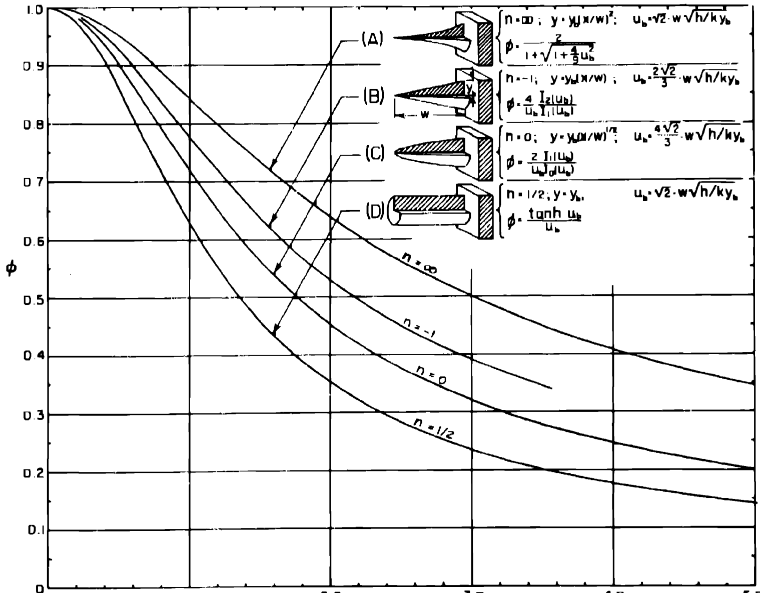

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Fig. 8. Efficiency Curves for Four Types of Spine

6. THEORY OF FLOW AND VISCOSITY

Friction Loss in Pipes

1. The following symbols are used with the meanings given here:

A = cross sectional area in sq ft
A_i = cross sectional area in sq in.
U = wetted perimeter in ft
U_i = wetted perimeter in in.
D = pipe diameter in ft
d = pipe diameter in in.
m' = hydraulic radius in ft
m = hydraulic radius in in.
l = length of pipe in ft
$g = 32.2$ fps² gravitational acceleration
P and ΔP = absolute pressure and pressure drop in psf
p and Δp = absolute pressure and pressure drop in psi
$r = P_1/P_2 > 1$ = pressure ratio
$\rho = 1/r = P_2/P_1 < 1$ = ratio of expansion
$\rho' = \gamma/g$ = mass density in slugs/cu ft
$\gamma = 62.4s$ = weight density in lb/cu ft
s = sp gr, relative to water
$V = 1/\gamma$ = specific volume in cu ft/lb
$\epsilon = (n-1)/n$ with $n = C_p/C_v$ the adiabatic exponent
v = velocity in fps
$h = v^2/2g$ = velocity head in ft

Flow

α = coefficient of discharge
G = flow in gpm; Q' in cfm; Q'' in cfs
$W' = Q' \cdot \gamma$ = flow in lb/min
W'' = flow in lb/sec

Head loss

ξ = general friction factor (for Darcy's equation)
$f = 4\xi$ friction factor for round pipes (Fanning's equation)
h' = head loss in ft, $\Delta p'$ = head loss in psi
h'_o and $\Delta p'_o$ the same for $l = 100$ ft of pipe
k = resistance factor

Viscosity

η = symbol for absolute viscosity:
η in poises $\qquad \eta_c$ in centipoises

$$\eta' \text{ in } \frac{\text{lb sec}}{\text{sq ft}} \qquad \eta'_h \text{ in } \frac{\text{lb hr}}{\text{sq ft}}$$

μ = symbol for dynamic viscosity:

$$\mu' \text{ in } \frac{\text{lb}}{\text{ft sec}} \qquad \mu'_h \text{ in } \frac{\text{lb}}{\text{ft hr}}$$

ν = symbol for kinematic viscosity:
$\nu = \eta/s$ in stokes
ν_c in centistokes
ν' in sq ft/sec
ν'_h in sq ft/hr

Flow and Velocity in Conduits

2. The average velocity in a conduit is defined as

$$v = Q''/A = 144 \cdot (Q''/A_i) \text{ fps}$$

With the flow in cfm or gpm and the pipe diameter in inches, we can write:

$$v = 144 \cdot (Q''/A_i) = 2.4 \cdot (Q'/A_i) = 0.321 \cdot (G/A_i) \text{ fps} \tag{1}$$

also

$$v = \frac{(183.35)}{d^2} \cdot Q'' = \frac{(3.056)}{d^2} \cdot Q' = \frac{(0.4085)}{d^2} \cdot G \text{ fps} \tag{1a}$$

If the flow is given as W' lb/min, we substitute $Q' = W'/\gamma = W'/62.4\ s$.

Another useful relation is

$$1 \text{ cfs} = 60 \text{ cfm} = 448.8 \text{ gpm}$$
$$1 \text{ gpm} = 500\ s \cdot \text{lb/hr}$$

Basic Energy Equation of Flow; Flow through Orifices and Nozzles

3. The velocity of any fluid is the result of an acceleration process during which a part of the fluid's potential energy (i.e., energy due to pressure) is changed into kinetic energy. The decrement in potential energy, which is the work spent on acceleration, appears as increment in kinetic energy:

$$V \cdot (-dP) = -dP/\gamma = d(v^2/2g) \ \frac{\text{ft lb}}{\text{lb}} \tag{2}$$

Liquids. In the case of liquids the work spent in accelerating the fluid from $v = 0$ velocity is:

$$V \cdot (P_1 - P_2) = \frac{\Delta P}{\gamma} = \frac{144 \cdot \Delta p}{\gamma} = \left(\frac{v^2}{2g}\right) \frac{\text{ft lb}}{\text{lb}} \tag{3}$$

since for liquids the specific volume $V = 1/\gamma$ remains constant.

The pressure drop along the path (concurrent with the acceleration) is numerically equal to the pressure at the bottom of a real or imaginary column of liquid standing h ft high above the point where the acceleration ends, and having a uniform density equal to that of the liquid at flow condition. This height

$$h = \Delta P/\gamma = \frac{144 \cdot \Delta p}{\gamma} = \left(\frac{v^2}{2g}\right) = \left(\frac{v}{8.025}\right)^2 \text{ft} \qquad (4$$

which is called the "velocity head," is identical with the height through which a free-falling body must drop to attain the velocity v fps.

Velocity of discharge. From Equation (4) we get

$$\begin{aligned} v &= \sqrt{2gh} = 8.025\sqrt{h} \text{ fps} \\ v &= 96.3\sqrt{\Delta p/\gamma} = 12.19\sqrt{\Delta p/s} \text{ fps} \end{aligned} \qquad (5)$$

as the theoretical velocity. Because of friction in the orifice or nozzle, the actual velocity is somewhat less.

Volume discharged. Inertia forces in the jet emerging from a plain, sharp-edged orifice cause the jet to contract, until it reaches its maximum velocity and minimum diameter in the so-called "vena contracta," where all the fluid particles of the jet move in parallel; thus the natural contours of this free jet conform to a converging nozzle.

The smallest cross section of the jet is only a fraction of the actual orifice area; the velocity in it is slightly less than the theoretical value. Both these facts are accounted for by applying a so-called coefficient of discharge α, the value of which depends on the quality of the orifice edge and also on the magnitude of the approach-velocity. For a truly sharp orifice with a negligible approach velocity $\alpha = 0.62$. Dulling the edge and increasing the approach velocity increases the value of α.

The jet from a nozzle has attained its minimum cross section inside the nozzle; thus there is no contraction of the jet after leaving the nozzle and the factor α accounts chiefly for the loss in velocity. The value of α is about .97 to .99, depending on the quality of the nozzle.

If the actual area of the orifice or nozzle is A_i sq in., the volume discharged will be

$$Q'' = \frac{\alpha \cdot A_i}{144} \cdot v \text{ cfs} \qquad (6)$$

with v in fps as in Equation (5).

In the more commonly used gpm units

$$\begin{aligned} G &= 25 \cdot (\alpha \cdot A_i) \cdot \sqrt{h} \text{ gpm} \\ G &= 300 \cdot (\alpha \cdot A_i) \cdot \sqrt{\Delta p/\gamma} \text{ gpm} \\ G &= 38 \cdot (\alpha \cdot A_i) \cdot \sqrt{\Delta p/s} \text{ gpm} \end{aligned} \qquad (7)$$

Weight discharged may be expressed as $W'' = Q'' \cdot \gamma$ lb/sec. More explicitly, we can write

$$\begin{aligned} W' &= 3.344\ (\alpha \cdot A_i)\gamma\sqrt{h} \text{ lb/min} \\ W' &= 40.125\ (\alpha \cdot A_i) \cdot \sqrt{\gamma \cdot \Delta p} \text{ lb/min} \\ W' &= 317\ (\alpha \cdot A_i) \cdot \sqrt{s \cdot \Delta p} \text{ lb/min} \end{aligned} \qquad (8)$$

In the equations given above the value of γ or s must be that existing at the actual conditions of the flow.

Gases and vapors. In the case of gases and vapors, a pressure drop is always accompanied by a corresponding increase of V, the specific volume; therefore the work spent on acceleration is

$$\left(\frac{v^2}{2g}\right) = \int_1^2 V \cdot (-dP) = P_1 \cdot V_1 \left[\frac{1-\rho^{\epsilon}}{\epsilon}\right] \text{ ft lb/lb} \qquad (9)$$

where $\rho = P_2/P_1$ is the expansion ratio and $\epsilon = (n-1)/n$, with n as the exponent in the $PV^n = $ constant equation for the adiabatic expansion line.

We should distinguish between three different cases:

If the **pressure drop is small** (less than 5% of the absolute pressure at the start), we can neglect the increase in volume and use the equations for liquids. We find the **velocity of discharge** to be

$$v = 96.3\sqrt{V_1 \Delta p} \text{ fps} \qquad (10)$$

which is identical with Equation (5), since $V = 1/\gamma$. (It is more convenient to use V than γ when dealing with gases.)

The **volume discharged** will be

$$Q' = 40.125\ (\alpha \cdot A_i) \cdot \sqrt{V_1 \cdot \Delta p} \text{ cfm} \qquad (11)$$

whereas the **weight discharged** will be

$$W' = 40.125\ (\alpha \cdot A_i) \cdot \sqrt{\Delta p/V_1} \text{ lb/min.} \qquad (12)$$

Equations (10) and (11) give a smaller

figure, while Equation (12) gives a larger one, than the true value. For a $\Delta p = 5\%$ pressure drop, the error in v and Q' will be about -1.25%, whereas the error in W' will be about $+3.75\%$. However, we can increase the accuracy of these equations substantially by multiplying Equations (10) and (11) by the factor $(1 + 0.25\Delta p/p_1)$, and Equation (12) by the factor $[1 - j \cdot (\Delta p/p_1)]$, where $\Delta p/p_1 = (1 - \rho)$ is the relative pressure drop and j is a factor varying with the value of n, as given in **Table 1.**

By applying these correction factors, we can extend the use of these equations almost as far as the critical pressure drop, with only a negligible error.

For a **medium pressure drop** ($p_2 > p_c$), we can either use the above equations with the proposed correction factors, or we can determine the correct values by means of the basic Equation (9) with $\rho = p_2/p_1 = (1 - \Delta p/p_1)$, finding the velocity to be

$$v = 96.3\sqrt{p_1 \cdot V_1} \cdot \left[\frac{1 - \rho^{\epsilon}}{\epsilon}\right]^{1/2} \text{ fps} \qquad (13)$$

and the flow to be

$$W' = 40.125 \ (\alpha \cdot A_i) \cdot \sqrt{p_1/V_1} \times \left[\rho^{1/n}\left(\frac{1 - \rho^{\epsilon}}{\epsilon}\right)^{1/2}\right] \text{ lb/min} \qquad (14)$$

For a fixed $p_1 - V_1$ starting condition, the flow W' varies, as does the value

$$b = \rho^{1/n}\left(\frac{1 - \rho^{\epsilon}}{\epsilon}\right)^{1/2}$$

which goes through a maximum at that critical expansion ratio, $\rho = \rho_c$, for which

$$\rho_c{}^{\epsilon} = \left(\frac{2}{n+1}\right) \qquad (15)$$

and

$$b \max = \beta = \left(\frac{n}{n+1}\right)^{1/2} \cdot \left(\frac{2}{n+1}\right)^{1/(n-1)} \qquad (16$$

both of which are only a function of n, the exponent for the adiabatic expansion line.

If in Equation (13) we substitute ρ^{ϵ} as in Equation (15), we find

$$v_c = 96.3\sqrt{\left(\frac{n}{n+1}\right) \cdot p_1 \cdot V_1} \text{ fps} \qquad (17)$$

as the velocity for the critical expansion ratio corresponding to the maximum flow

$$W_c' = 40.125 \ (\alpha \cdot A_i) \cdot \beta\sqrt{p_1/V_1} \text{ lb/min} \qquad (18)$$

with β as in Equation (16).

The corresponding values of β, ρ_c and $\epsilon = (n-1)/n$, for all feasible n values, are given in Table 1.

If the **pressure drop is large,** during the acceleration process the velocity, v, of the gas increases steadily, as does its volume, V. The **rate** of the velocity increase is greatest at the very start, diminishing as expansion progresses. The contrary is true for the rate of the volume increase. As a result the ratio V/v, which is a measure of the nozzle cross-section required, first decreases as expansion progresses, reaching its minimum at the critical expansion ratio in the vena contracta. After that it increases, indicating that the nozzle cross-section also must widen out, in order to accommodate the increasing volume as expansion proceeds (Laval nozzle). If such widening of the nozzle is not provided for,

Table 1. Data for Gas Flow

n	$1/n$	$\epsilon = \frac{n-1}{n}$	j	$r_c = 1/\rho_c$	ρ_c	$\left(\frac{\Delta p}{p_1}\right)_c$	β
1.00	1.	0	.775	1.649	.6065	.3935	.429
1.05	.9524	.0476	.741	1.679	.5955	.4045	.4365
1.10	.9091	.0909	.710	1.709	.5850	.4150	.4440
1.15	.8696	.1304	.682	1.741	.5745	.4255	.4513
1.20	.8333	.1667	.655	1.771	.5645	.4355	.4584
1.25	.8000	.2000	.631	1.802	.5550	.4450	.4652
1.30	.7692	.2308	.608	1.832	.5460	.4540	.4717
1.35	.7407	.2593	.587	1.862	.5370	.4630	.4780
1.40	.7143	.2857	.567	1.894	.5280	.4720	.4840
1.45	.6896	.3104	.549	1.923	.5200	.4800	.4900
1.50	.6667	.3333	.532	1.953	.5120	.4880	.4960

Table 2. Viscosity of Freons and Methyl Chloride in Centipoises (η_c)

t F	Liquids at saturation pressure							Vapors at 1 atm						
	F-11	F-12	F-21	F-22	F-113	F-114	Methyl chloride	F-11	F-12	F-21	F-22	F-113	F-114	Methyl chloride
−40	.980	.423	.629	.351		.879	.349	.0088	.0106	.0094	.0105		.0095	.0086
−20	.801	.371	.547	.316	1.566	.711	.321	.0092	.0109	.0098	.0109	.0090	.0098	.0090
0	.677	.335	.484	.291	1.263	.598	.298	.0096	.0113	.0101	.0113	.0093	.0102	.0094
+20	.586	.308	.436	.271	1.043	.516	.279	.0099	.0116	.0105	.0118	.0095	.0106	.0098
40	.517	.286	.397	.256	.876	.454	.263	.0103	.0119	.0108	.0122	.0098	.0109	.0101
60	.461	.269	.364	.243	.747	.405	.249	.0106	.0123	.0111	.0126	.0101	.0112	.0105
80	.417	.255	.337	.232	.646	.366	.237	.0110	.0126	.0115	.0130	.0103	.0116	.0108
100	.380	.242	.314	.223	.564	.334	.226	.0113	.0129	.0118	.0133	.0106	.0119	.0111
120	.349	.232	.294	.214	.497	.307	.217	.0116	.0132	.0121	.0137	.0108	.0122	.0115
140	.323	.222	.277	.207	.442	.284	.208	.0120	.0135	.0124	.0141	.0111	.0125	.0118
160	.300	.214	.262	.201	.395		.200	.0123	.0138	.0127	.0145	.0113		.0122
180	.281	.207	.248	.195	.356		.193	.0126	.0140	.0130	.0148	.0116		.0125
200	.263	.200	.236		.322		.186	.0129	.0143	.0133	.0152	.0118		.0128
220	.248		.225		.293		.180	.0132	.0146	.0136	.0156	.0120		.0131
240	.232		.215		.268		.175	.0135	.0149	.0139	.0159	.0123		.0134

Data by Benning & Markwood, *Refrig. Eng.* April 1939.

the gas cannot expand beyond the critical ratio. Thus the velocity given in Equation (17) is the maximum attainable with converging nozzles or plain orifices.

After the jet emerges from the nozzle or the vena contracta, the gas is free to expand further. However, unguided by surrounding walls, the pressure drives in every direction, the motion becomes irregular, and the jet explodes.

The flow, as given in Equation (18), will be the maximum in all cases, even if the nozzle widens out, since widening the nozzle increases the velocity but not the flow, which is fixed by the cross sectional area of the throat.

Theory of Viscosity

4. Viscosity is the cause of the fluid friction, which appears whenever two adjacent layers of a fluid slide past each other. This frictional resistance is of the nature of a shearing stress, its magnitude depending on the rate of sliding as well as on the viscosity. For one square foot of sliding area the drag is

$$\tau = \eta' \cdot (dv/dx) \text{ psf} \qquad (19)$$

where η' is the absolute viscosity and dv/dx is the velocity variation normally across one unit thickness of the sliding layer in fps/ft = 1/sec units, (called velocity gradient). From Equation (19), the magnitude and dimension of the absolute viscosity in fps units is

$$\eta' = \frac{\tau}{(dv/dx)} \text{ in } \frac{\text{lb sec}}{\text{sq ft}} \text{ units} \qquad (20)$$

In the scientific literature the absolute viscosity is given in "poises" (i.e., in dyne sec /sq cm units) and is designated by η; often it is given in centipoises (1/100th of a poise), and is designated by η_c (also by Z). Since 1 lb $= 981 \times 453.6$ dyne and 1 sq ft $= 929$ sq cm, it is evident that one (lb sec /sq ft) unit = 479 poises. Thus η poises or η_c centipoises are equal to

$$\eta/479 = \eta_c/47900 = \eta' \text{ in } \frac{\text{lb sec}}{\text{sq ft}} \qquad (21)$$

The **Reynolds number,** *Re*, is an important criterion when analyzing flow problems. For round pipes,

$$Re = \frac{D \cdot v \cdot \rho}{\eta'} \qquad (22)$$

which is dimensionless and without any factor, if all components are given in consistent units. In this formula ρ is the mass-density (i.e., $\rho = \gamma/g$). Substituting this we get

$$Re = \frac{D \cdot v \cdot \gamma}{(\eta' \cdot g)} = \frac{D \cdot v \cdot \gamma}{\mu'} \qquad (22a)$$

This value $\eta' \cdot g = \mu'$ also is called viscosity, although it appears in lb/ft sec units. Often the two units are badly confused in engineering literature. For the sake of clarity let us call this unit "dynamic" viscosity. Its numerical relation with the poise and centipoise unit is

$$\mu' = 32.2\eta' = 32.2(\eta/479)$$

$$= \frac{\eta}{14.88} = \frac{\eta_c}{1488} \text{ in } \frac{\text{lb}}{\text{ft sec}} \tag{23}$$

The ratio $\eta'/\rho' = \mu'/\gamma$ is called "kinematic" viscosity. Its dimension is sq ft/sec, and it is designated by ν'. With $\gamma = 62.4\ s$, from Equation (23) we derive

$$\nu' = \mu'/\gamma = \frac{\mu'}{62.4s} = \frac{(\eta/s)}{929} \tag{24}$$

$$= \nu/929 \text{ in sq ft/sec}$$

where $(\eta/s) = \nu$ is the kinematic viscosity in sq cm/sec units, called "stokes." Analogously, the ratio $\eta_c/s = \nu_c$ is called centistoke (1/100th of the stoke).

In the study of heat transfer the viscosity units are based on the hour as the unit of time. With η the absolute viscosity in poises we find:

$$\eta'_h = \frac{\eta'}{3600} = \frac{\eta}{1.725 . 10^6} \text{ in } \frac{\text{lb hr}}{\text{sq ft}} \text{ units}$$

$$\mu'_h = 3600 \cdot \mu' = 242 \cdot \eta \text{ in } \frac{\text{lb}}{\text{ft hr}} \text{ units}$$

$$\nu'_h = 3600 \cdot \nu' = 3.875\ (\eta/s) \text{ in } \frac{\text{sq ft}}{\text{hr}} \text{ units}$$

Thus we have three types of units for expressing viscosity: (1) abs unit = force · time /area (symbol η); (2) dynamic unit = force /length · time (symbol μ); and (3) kinematic unit = area/time (symbol ν).

When making computations involving viscosity, it is important to distinguish between these units in order to avoid utter chaos in our calculations.

The viscosity of liquids decreases rapidly with rising temperature, and vice versa. As a rough rule we may say that the logarithm of the absolute viscosity varies linearly with the reciprocal of the absolute temperature; that $\log \eta = a + b/T$, is a straight line in a semilog plot with η on the logarithmic scale.

The viscosity of gases, on the contrary, increases with the temperature.

Viscous Flow

5. The magnitude of the Reynolds number is a criterion of the type of flow there

Table 3. Viscosity of Liquid NH_3, Centipoises (η_c)

Pressure, psia	Temperature, F							Saturation	
	0	10	20	30	40	50	60	η_c	temp, F
30	—	—	—	—	—	—	—	.2503	—.6
40	.2539	.2482	—	—	—	—	—	.2474	11.7
50	.2562	.2509	.2444	—	—	—	—	.2432	21.7
60	.2585	.2533	.2474	.2398	—	—	—	.2398	30.2
70	.2605	.2556	.2499	.2427	—	—	—	.2362	37.7
80	.2624	.2576	.2521	.2553	.2369	—		.2328	44.4
90	.2643	.2595	.2541	.2476	.2397	.2301	—	.2295	50.5
100	.2661	.2613	.2559	.2496	.2420	.2331	—	.2264	56.0
120	.2691	.2645	.2594	.2534	.2461	.2376	.2277	.2204	66.0
140	.2718	.2675	.2625	.2567	.2495	.2413	.2318		
160	.2740	.2699	.2651	.2594	.2526	.2445	.2355		
180	.2760	.2720	.2674	.2617	.2549	.2472	.2387		
200	.2778	.2740	.2695	.2639	.2571	.2496	.2510		
220	.2794	.2756	.2712	.2657	.2590	.2515	.2430		
240	.2808	.2770	.2727	.2673	.2607	.2532	.2447		
260	.2820	.2783	.2740	.2687	.2623	.2542	.2463		
280	.2833	.2796	.2752	.2699	.2636	.2560	.2477		
300	.2844	.2807	.2763	.2710	.2647	.2572	.2489		
320	.2853	.2817	.2773	.2719	.2657	.2584	.2501		
340	.2861	.2825	.2782	.2728	.2666	.2594	.2510		
360	.2868	.2833	.2789	.2736	.2675	.2603	.2520		

Table 4. Viscosity of Liquid Methyl Chloride, Centipoises (η_c)

Pressure, psia	Temperature, F									Saturated liquid	
	0	10	20	30	40	50	60	70	80	η_c	temp, F
20	.3083	—	—	—	—	—	—	—	—	.3064	2.9
30	.3133	.3064	.2993	—	—	—	—	—	—	.2975	22.3
40	.3177	.3112	.3043	.2965	—	—	—	—	—	.2901	36.6
50	.3210	.3145	.3079	.3002	.2911	—	—	—	—	.2826	48.4
60	.3239	.3176	.3109	.3033	.2943	.2846	—	—	—	.2760	58.6
70	.3265	.3202	.3136	.3061	.2971	.2877	.2778	—	—	.2704	67.2
80	.3283	.3222	.3154	.3079	.2994	.2903	.2805	.2701	—	.2634	76.1
90	.3297	.3238	.3170	.3095	.3011	.2921	.2824	.2719	.2607	.2581	82.2
100	.3303	.3248	.3181	.3107	.3023	.2933	.2835	.2731	.2617	.2505	88.7

will be in conduit. If $Re<1200$, the flow is always viscous; this flow is characterized by the fact that the fluid particles move past each other in parallel lines without mixing. The velocity distribution over the cross section depends on the shape of the conduit.

In round pipes viscous flow is coexistent with a parabolic velocity distribution over the cross section. The maximum velocity in the pipe center is twice the average velocity expressed as $v=Q''/A$. Right at the pipe surface the velocity is nil, but at that point the velocity gradient (dv/dx) has its maximum value of $8v/D$. The force required to move the contents of an l-ft long straight pipe against this fluid friction at the wall is

$$\Delta P\cdot A = U\cdot l\cdot \eta'\cdot (dv/dx) = 8\pi\eta' vl \text{ lb} \qquad (25)$$

with ΔP as the pressure drop in psf over the l ft of pipe and $(dv/dx)=8v/D$ right at the inside surface.

With the pipe diameter in inches, the viscosity in poises, and the flow in gpm or cfm, we find the pressure drop for $l=100$ ft in psi as

$$\Delta p'_0 = \frac{6.68v\eta}{d^2} = \frac{2.73G\eta}{d^4} = \frac{20.42Q'\eta}{d^4} \text{ psi} \qquad (26)$$

or in ft of the fluid flowing

$$h'_0 = \frac{15.42v\nu}{d^2} = \frac{6.3G\nu}{d^4} = \frac{47.11Q'\nu}{d^4} \text{ ft} \qquad (27)$$

where $\nu=\eta/s$ is the kinematic viscosity of the fluid in stokes with η and s for the condition existing at the flow.

(If the viscosity is taken in centipoises ($\eta_c=100\eta$), or in centistokes ($\nu_c=100\nu$), Equations (26) and (27) must be divided by 100.)

Equation (27) is used with some modifications for determining ν, the kinematic viscosity of fluids. The most common method for this determination is to meas-

Table 5. Viscosity of Liquid Sulfur Dioxide, Centipoises (η_c)

Pressure, psia	Temperature, F								Saturated liquid	
	0	10	20	30	40	50	60	70	η_c	temp, F
10	—	—	—	—	—	—	—	—	.4790	−1.34
20	.4840	.4544	.4232	—	—	—	—	—	.4013	26.44
30	.4921	.4629	.4324	.4003	—	—	—	—	.3470	44.75
40	.4980	.4695	.4393	.4074	.3735	.3366	—	—	.3040	58.33
50	.5028	.4747	.4451	.4134	.3792	.3431	.3050	.2665	.2548	70.40
60	.5065	.4794	.4499	.4184	.3841	.3484	.3108	.2731	—	—
70	.5099	.4834	.4541	.4225	.3885	.3529	.3159	.2789		
80	.5125	.4868	.4577	.4261	.3921	.3564	.3198	.2835		
90	.5147	.4894	.4609	.4292	.3950	.3595	.3233	.2874		
100	.5165	.4917	.4634	.4319	.3975	.3621	.3265	.2909		
110	.5180	.4937	.4655	.4340	.3997	.3643	.3293	.2940		

Table 6. Viscosity of Refrigerants in Centipoises (η_c)

t C	NH$_3$			CO$_2$			SO$_2$			Butane	Propane	Ethane	Ethylene	t C
	Liquid	Vapor		Liquid	Vapor		Liquid	Vapor		Vapor at 1 atm				
	at ps	at 1 atm	at ps	at ps	at 1 atm	at ps	at ps	at 1 atm	at ps					
−60	.380	.0071			.0108								.00735	−60
−40	.275	.0078			.0118								.00805	−40
−20	.255	.0086	.0109	.120	.0128		.465		.0106				.00875	−20
−10	.245		.0113	.111		.0167	.412		.0113					−10
0	.240	.0093	.0118	.087	.0138	.0174	.368	.0116	.0123	.0069	.0075	.0086	.00940	0
+10	.230		.0124	.071		.0183	.333		.0135					+10
20	.220	.0100	.0129	.048	.0148	.0203	.304	.0126	.0151	.0074	.0081	.0092	.01005	20
30				.032		.0235	.279		.0169					30
50		.0111			.0162			.0140		.0082	.0088	.0101	.01100	50
100		.0130			.0185			.0163		.0095	.0101	.0115	.01260	100
150		.0148			.0208			.0186		.0110	.0113	.0128	.01400	150
200		.0166			.0229			.0207			.0125	.0142	.0154	200
250		.0184			.0249			.0227			.0136	.0154	.0166	250
300		.0202			.0268			.0246			.0144			300

Data from Henning, Wärmetechnische Richtwerte. VDI—1938.

Table 7. Kinematic Viscosity of Water (ν_c in centistokes)

t, F	ν_c	t, F	ν_c	t, F	ν_c	t, F	ν_c
32	1.794	80	.865	130	.517	180	.357
40	1.546	90	.768	140	.478	190	.336
50	1.310	100	.687	150	.441	200	.316
60	1.130	110	.622	160	.410	210	.299
70	.984	120	.565	170	.383	220	.283

Data from Smithsonian tables.

ure the time (t) as the number of seconds required by a fixed volume of the fluid to pass by gravity through a capillary of strictly fixed dimensions. The time, found with the Saybolt universal viscometer, is expressed as "Universal Saybolt seconds." The relation between this time and the kinematic viscosity in centistokes is

$$\nu_c = \eta_c/s = \left(0.220t - \frac{180}{t}\right) \text{ centistokes} \qquad (28)$$

(For $\nu_c = 1$, the kinematic viscosity of water at 68.4 F, the Saybolt time is 31 seconds.)

From the discussion above, it becomes apparent that citing viscosity data without the corresponding temperatures is meaningless.

Turbulent Flow and Reynolds Number (*Re*)

6. Liquids. Whenever the Reynolds number $Re > 2500$, the flow will be turbulent; the frictional resistance increases almost with the square of the velocity as compared to the linear increase during viscous flow.

The basic relation for the head loss from frictional resistance is Darcy's equation

$$h' = \xi \cdot l/m' \cdot \left(\frac{v^2}{2g}\right) \text{ ft} \qquad (29)$$

which gives the head loss as a multiple of the velocity head in feet of the fluid flowing. In this equation, ξ is the general friction factor (for any type of conduit), and $m' = (A/U)$ is the hydraulic radius in ft for the conduit in question.

For round pipes $m' = D/4$. Thus $\xi/m' = 4\xi/D = f/D$ and we get

$$h' = f \cdot \frac{l}{D} \cdot \left(\frac{v^2}{2g}\right) = f \cdot \frac{12l}{d} \cdot \left(\frac{v^2}{2g}\right) \text{ ft} \qquad (30)$$

which is called **Fanning's equation**, and is useful only for round pipes with $f = 4\xi$ as the friction factor.

It is customary to write up the friction loss equation for $l = 100$ ft of actual of equivalent length of pipe. With the hydraulic radius as $m=(A_i/U_i)$ in inches, and the diameter also in inches, by combining all constants into one factor we get

$$h'_0=18.63\ (\xi/m)\cdot v^2=18.63\ (f/d)\cdot v^2 \text{ ft} \quad (31)$$

or

$$\begin{aligned}\Delta p'_0&=8.075s\cdot(\xi/m)\cdot v^2\\&=8.075s\cdot(f/d)\cdot v^2 \text{ psi}\end{aligned} \quad (32)$$

When selecting a proper value for the friction factor, care must be taken to determine whether it is a ξ value or an $f=4\xi$ value, and to use the proper equation.

With the flow in gpm, cfm, or in lb/min, we can modify Fanning's equation for $l=100$ ft of round pipe into

$$h'_0=3.11f\cdot\frac{G^2}{d^5}=174f\cdot\frac{(Q')^2}{d^5} \text{ ft} \quad (33)$$

or

$$\Delta p'_0=1.35fs\cdot\frac{G^2}{d^5}=75.4fs\cdot\frac{(Q')^2}{d^5} \text{ psi} \quad (34)$$

and

$$\Delta p'_0=1.208f\cdot\frac{(W')^2}{\gamma\cdot d^5}=\frac{1.936}{100}\cdot f\cdot\frac{(W')^2}{s\cdot d^5} \text{ psi} \quad (35)$$

By coordinating the results of a great many tests of turbulent flow, it has been found that the friction factor can be plotted as a function of the Reynolds number Re. It is usually plotted on a logarithmic scale.

McAdams gives such a plot; the lower line is for drawn, smooth tubes, whereas the upper line gives good average values for standard commercial steel pipes.

This empirical $\xi=f(Re)$ relation can be expressed also by an equation; the values for smooth tubes follow the equation

$$1000\xi=1.4+125/Re^{0.32}$$

whereas those for standard steel pipes are

$$1000\xi=3.07+188.6/Re^{0.38}$$

Kemler and Pigott have also analyzed a great number of tests and segregated the friction factor data into different pipe diameter groups, which they plotted separately against Re, the Reynolds number. These plots reflect the influence of the relative roughness of the pipe on the magnitude of the friction factor.

Moody has correlated Reynolds number, diameter, roughness and friction factor by means of the Colebrook function,

$$\frac{1}{\sqrt{f}}=-2\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{R\sqrt{f}}\right)$$

where ε is a factor, expressed in feet, representative of the roughness of the pipe, which for any type of pipe is constant irrespective of diameter. Stickney has expressed this in chart form, **Fig. 1,** in which read from diameter at the top, down to type of pipe, across to Re, and interpolate for f. Note that $f=4\xi$.

When figuring a case of pipe friction loss, one must know the numerical magnitude of all the variables given in the equations. With these figures known, one first calculates the Reynolds number Re pertaining to the specific case. This value of Re not only determines the magnitude of the friction factor, but it is also the criterion of what type of flow will exist. If $Re<1200$, the flow is definitely viscous; if $Re>2500$, it will be turbulent. For Re values in between, the flow is uncertain.

While the above equations serve primarily for finding the friction loss in turbulent flow, they can be used also for viscous flow simply by substituting $\xi=16/Re$, or $f=64/Re$.

The Reynolds number, which obviously plays an important role, can be expressed in different ways. Its basic form is given by Equations (22) and (22a); other forms are (with η in poises and $\nu=(\eta/s)$ in stokes)

$$Re=1.24\ \frac{dv\gamma}{\eta}=77.42\ \frac{dv}{\nu} \quad (36)$$

and with the flow in gpm, cfm or lb/min,

$$\begin{aligned}Re&=0.5066\ \frac{G\cdot\gamma}{d\cdot\eta}=31.62\ \frac{G}{d\cdot\nu}\\Re&=3.79\cdot\frac{Q'\cdot\gamma}{d\cdot\eta}=236.5\ \frac{Q'}{d\cdot\nu}\\Re&=3.79\ \frac{W'}{d\cdot\eta}\end{aligned} \quad (37)$$

(If the viscosity is taken in centipoises ($\eta_c=100\eta$) or in centistokes ($\nu_c=100\nu$), then Equations (36) and (37) must be multiplied by 100 to get Re correctly.)

For "not-round" ducts, we substitute in Equation (36), but not in Equation (37).

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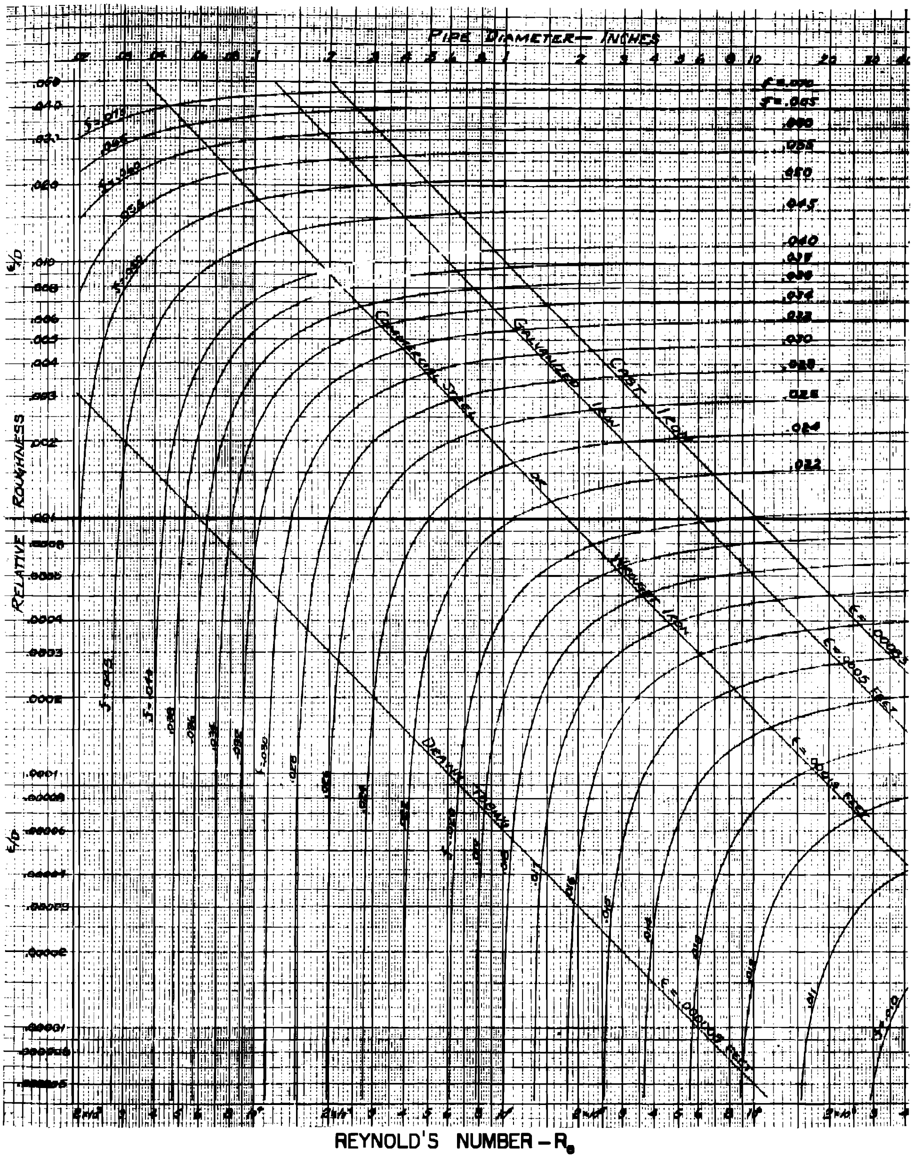
[1] These are ξ values; multiply by four to get f values.



f = FRICTION FACTOR
($f = 4\xi$)

$\frac{\varepsilon}{D}$ = RELATIVE ROUGHNESS
ε AND D IN SAME UNITS (inches, feet)

ε = ABSOLUTE ROUGHNESS

D = PIPE DIAMETER

$$R_e = \frac{VD}{\nu'} = 7740\frac{\delta V}{\nu_s} = 3162\frac{G}{\delta \nu_c}$$

V = FLUID VELOCITY ft/sec

δ = PIPE DIAMETER inches

D = " " feet

ν' = KINEMATIC VISCOSITY sq ft/sec

ν_c = " " Centistoke

G = GALLONS PER MINUTE

Fig. 1. Friction Factors

the diameter with its equivalent in terms of hydraulic radius: $d = 4m = (A_i/U_i)$, with the velocity figured according to Equation (1).

For annular cross sections (double-pipe heat exchangers) $m = (d_1 - d_2)/4$, with d_1 the inside diameter of the outside pipe and d_2 the outside diameter of the inside pipe in inches.

Gases and vapors. All our formulae for friction loss hold good to a limited degree also for gas and vapor lines, provided the pressure drop is less than 10% of the initial absolute pressure. Most friction loss problems encountered by the refrigerating engineer in connection with gas and vapor lines fall well within this restriction, and thus can be handled with the simple equations for liquids. However, for easier handling one may substitute $\gamma = 1/V$

$$\begin{aligned}\Delta p'_0 &= .1294f\,\frac{v^2}{V \cdot d} = 1.208f \cdot \frac{(Q')^2}{V \cdot d^5} \\ &= 1.208f \cdot \frac{V \cdot (W')^2}{d^5}\ \text{psi}\end{aligned} \tag{38}$$

for $l = 100$ ft of pipe, with $V = 1/\gamma$ as the specific gas volume in cu ft/lb (for the p_1 starting pressure).

If greater accuracy is desired, one may first figure a preliminary $\Delta p'$ value, and then with this calculate the correction factor $(1 + \Delta p'/2 \cdot p_1)$ as a multiplier for the erstwhile $\Delta p'$.

The friction factor f in Equation (38) is a function of the Reynolds number *Re*. McAdams and Sherwood found that for air, gas, and steam flow one may write

$$f = 4\xi = 0.0216 + 186/Re \tag{39}$$

with *Re* as per any of its equations.

However, for gas flow the most convenient procedure is to express *Re* with the flow as W' in lb/min. Since for gases η varies only with the temperature, *Re* and f also remain constant during the flow, if the temperature stays constant. Both the velocity and gas density change during the flow, but in such a manner that $v \cdot \gamma = v/V$ remains constant.

In his many tests with air, gas, and saturated and superheated steam, Fritzsche found that the friction factor f, primarily a function of the fluid weight passing through the pipe, depends only slightly on the diameter. His test results can be expressed as

$$f = \frac{3.30}{100} \cdot \frac{d^{.027}}{(W')^{.143}} = \frac{3.45}{100 \cdot (W')^{.143}} \tag{40}$$

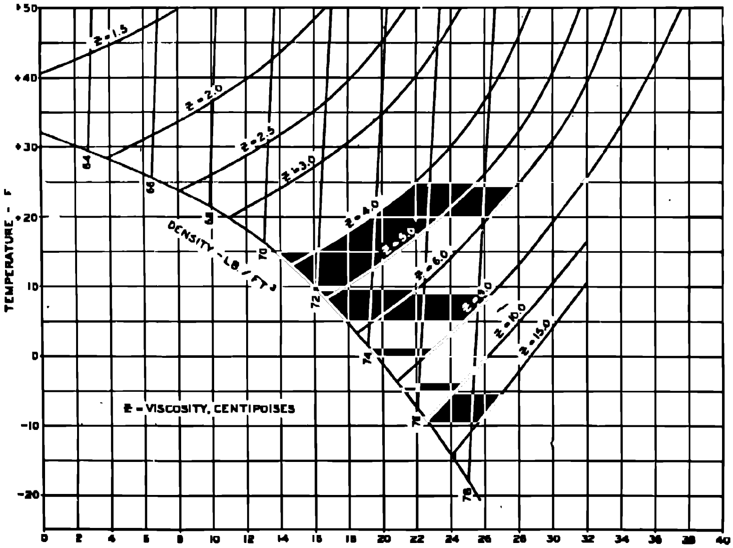


Fig. 2. Diagram of Viscosity Values of Calcium Chloride Brine

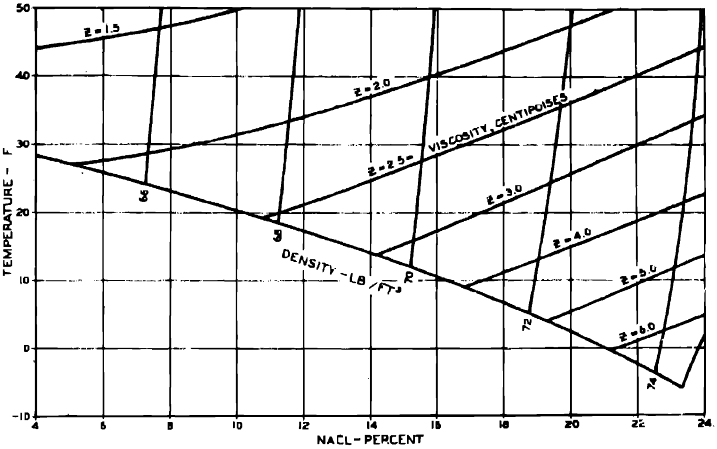


Fig. 3. Viscosity and Density of Sodium Chloride

The left side of Equation (40) is the original equation, while the right side is a simplification; since $d^{.027}$ varies only slightly with d, an average value, say $5^{.027} = 1.0444$, can be substituted for it, resulting in 3.45, as shown above. This equation of Fritzsche, which may be used very conveniently, gives f values close to those found in Equation (39).

Miscellaneous Losses

7. Losses due to obstructions in the line. Elbows, fittings, and valves in the line disturb the flow pattern, causing an additional head loss, which can be expressed either by a resistance factor k (giving this head loss as a multiple of the velocity head: $h' = k(v^2/2g)$ ft), or by an equivalent length of pipe (l_e) of the same diameter, which may be added to the actual pipe length when figuring the total friction loss.

The Crane Company, in its "Flow of Fluids," has published a nomogram (Fig. 6) which gives the equivalent length for almost any available fitting.

The resistance of **bends** and **welding elbows** is less than that of screw elbows, depending on the (R/d) relative radius of the bend; 90° bends show a minimum of resistance, when (R/d) is around 3. The equivalent length for 90° bends, expressed in pipe diameters, are given in the Tube-Turns[11] catalog as follows:

R/d	l_e (in. diam)
.5	40
1.0	18.5
1.5	12.2
2	10
2.5	9.2
3	9
3.5	9.2
4	10
4.5	11

For 45° bends take 64% and for 180° bends 134% of the given l_e values.

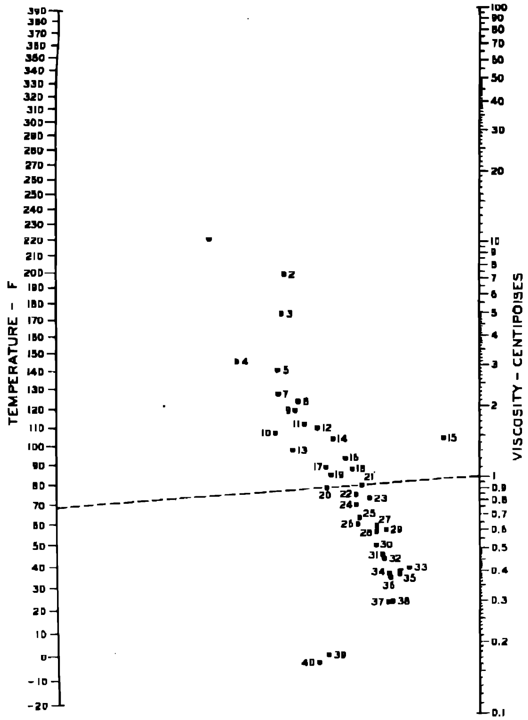
The resistance of **single-weld mitre bends** varies with δ, the angle of the deviation. According to Kirchbach (Hutte: 26th ed):

δ	22.5°	30°	45°	60°	90°
k	.11	.17	.32	.68	1.27

Where k is a multiplying factor of $v^2/2g$.

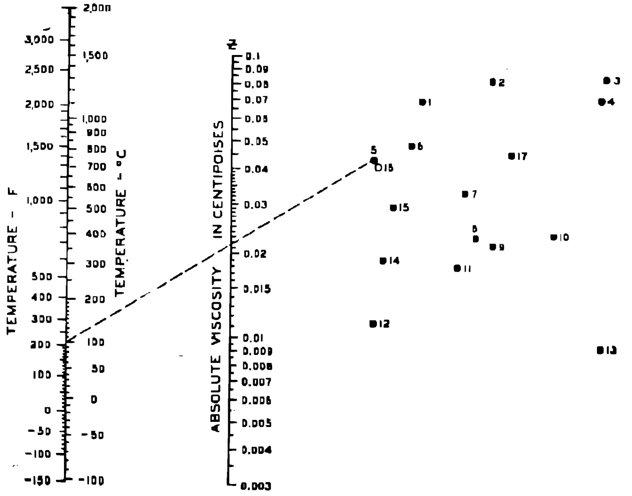
Entrance loss. Whenever a fluid enters a pipe and the jet suffers a contraction at the entrance, there will be a loss of head. The jet is first speeded up to the greater velocity v_0 in the vena contracta, from which it slows down again to the lesser velocity v in the pipe; during this slowdown the kinetic energy of the jet is never fully reconverted into static head (pressure), since part of it is lost in impact, turbulence and eddies.

The head consumed in the speed-up process is $h_0 = v_0^2/2g = (1+k) \cdot v^2/2g$ ft, of which $v^2/2g$ remains in the fluid and



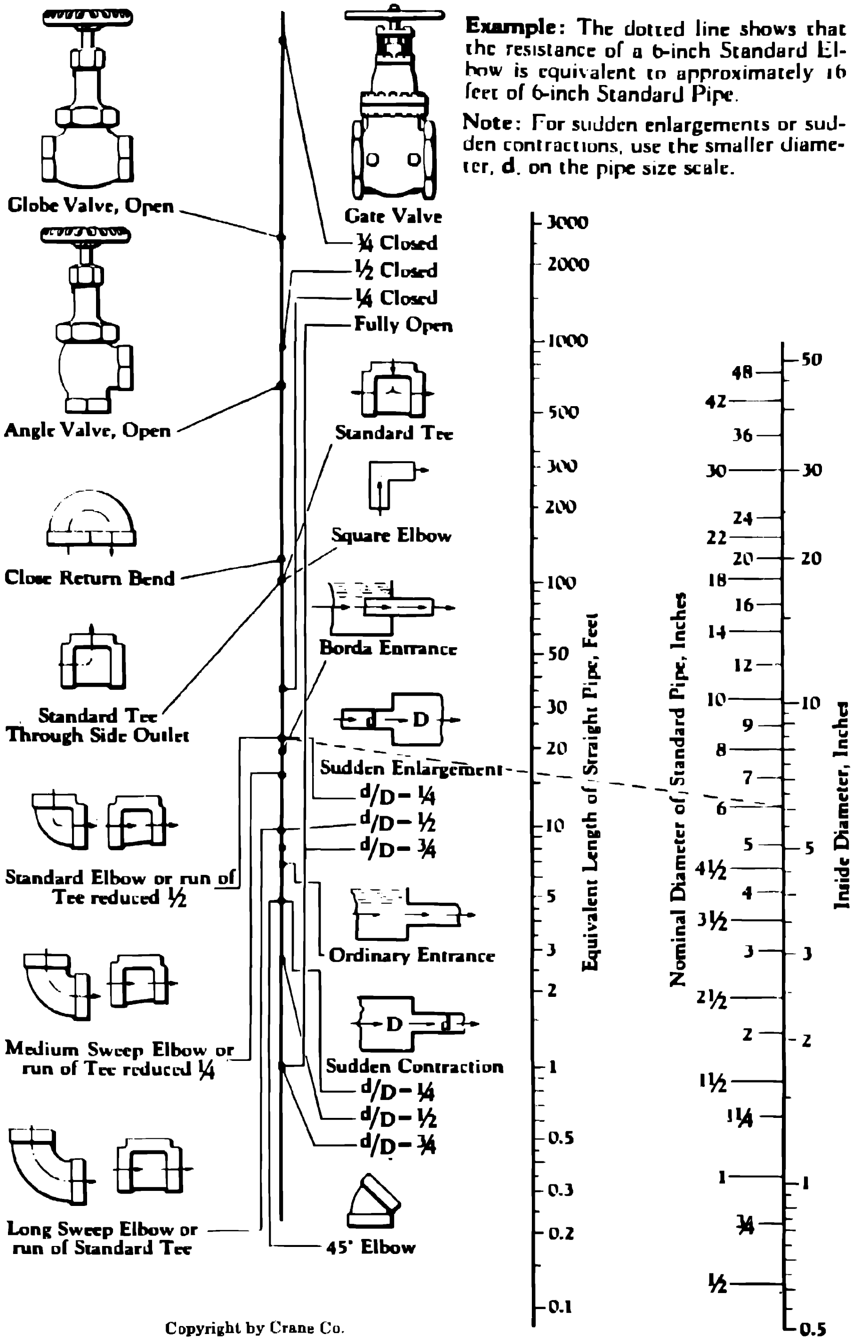
18	Acetic Acid 100 %
12	Acetic Acid 70 %
34	Acetone
17	Allyl Alcohol
39	Ammonia
8	Aniline
26	Benzene
11	Butyl Alcohol
40	Carbon Dioxide
33	Carbon Disulfide
21	Carbon Tetrachloride
22	Chlorbenzene
29	Chloroform
37	Ether
19	Ethyl Alcohol 100 %
10	Ethyl Alcohol 40 %
30	Ethyl Acetate
23	Ethylene Chloride
4	Ethylene Glycol
1	Glycerol 100 %
7	Glycerol 50 %
31	Heptane
36	Hexane (N)
15	Mercury
26	Methyl Alcohol 100 %
24	Methyl Alcohol 90 %
13	Methyl Alcohol 30 %
32	Methyl Acetate
9	Naphthalene
14	Nitrobenzene
28	Octane
38	Pentane
5	Phenol
35	Sulfur Dioxide
2	Sulfuric Acid 111 %
3	Sulfuric Acid 98 %
6	Sulfuric Acid 60 %
27	Toluene
16	Turpentine
20	Water

Fig. 4. Viscosity of Various Liquids[6]



1—O_2
2—NO
3—CO_2
4—HCl
5—Air
6—N_2
7—SO_2
8—CH_4
9—H_2O
10—NH_3
11—C_2H_6
12—H_2
13—C_6H_6
14—$9H_2+1N_2$
15—$3H_2+1N_2$
16—CO
17—Cl_2

Fig. 5. Viscosity of Gases at Atmospheric Pressure[6]



Example: The dotted line shows that the resistance of a 6-inch Standard Elbow is equivalent to approximately 16 feet of 6-inch Standard Pipe.

Note: For sudden enlargements or sudden contractions, use the smaller diameter, **d**, on the pipe size scale.

Copyright by Crane Co.

Fig. 6. Resistance of Valves and Fittings to Flow of Fluids

$k \cdot v^2/2g$ is lost. The greater the contraction of the jet, the larger is k.

If the **pipe inlet is flush** with the face of the wall and well rounded, there will be no contraction, and thus no entrance loss: $k = .06$ or less and covers only the friction loss proper. If the pipe edge is only slightly rounded, $k = 0.25$; and if the edge is a sharp 90°, then $k = 0.50$.

If the **pipe protrudes** beyond the face of the wall (Borda entrance), the contraction increases, and thus the k value is increased. For standard pipes with a sharp 90° inside edge, $k = 0.80$.

The so-called **"reversal loss"** in water-boxes of multipass coolers and condensers is the same $h_0 = (1 + k) \cdot v^2/2g = h_r'$ ft head loss for each pass; the fluid must be re-accelerated during its entrance to each successive pass. The Heat Exchange Institute has published data for reversal losses which follow the equation

$$h_r' = \frac{8.35}{100} \cdot v^{1.48} \text{ ft} \qquad (41)$$

for each pass with v the velocity in the tubes in fps.

Loss from sudden enlargements. In this case the fluid jet, going from the smaller pipe with the velocity v_1 into the larger one, plunges into a turbulent mass of fluid, from which it finally emerges with the velocity v_2 fixed by the flow and the pipe size. The loss of energy during this impact is expressed by the Borda-Carnot equation as

$$h' = (v_1 - v_2)^2/2g = \frac{v_1^2}{2g}\left(1 - \frac{v_2}{v_1}\right)^2 \text{ ft}$$

To avoid head loss in pipe transitions, they should be made as slender cones with an included angle of not more than 8° to 10° and with smooth joints.

Loss from sudden contraction. This is essentially of the same nature as the entrance loss. However, the fluid approaching the reduced inlet already has the velocity v_1. Thus the contraction at the new entrance is less and k correspondingly less, the larger the velocity ratio v_1/v_2, with v_2 as the velocity in the following smaller pipe. O'Brien and Hickox give the following data:

v_1/v_2	k
0.1	.362
0.2	.338
0.3	.308
0.4	.267
0.5	.221
0.6	.164
0.7	.105
0.8	.053
0.9	.015

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Methyl Formate
Methylene Chloride

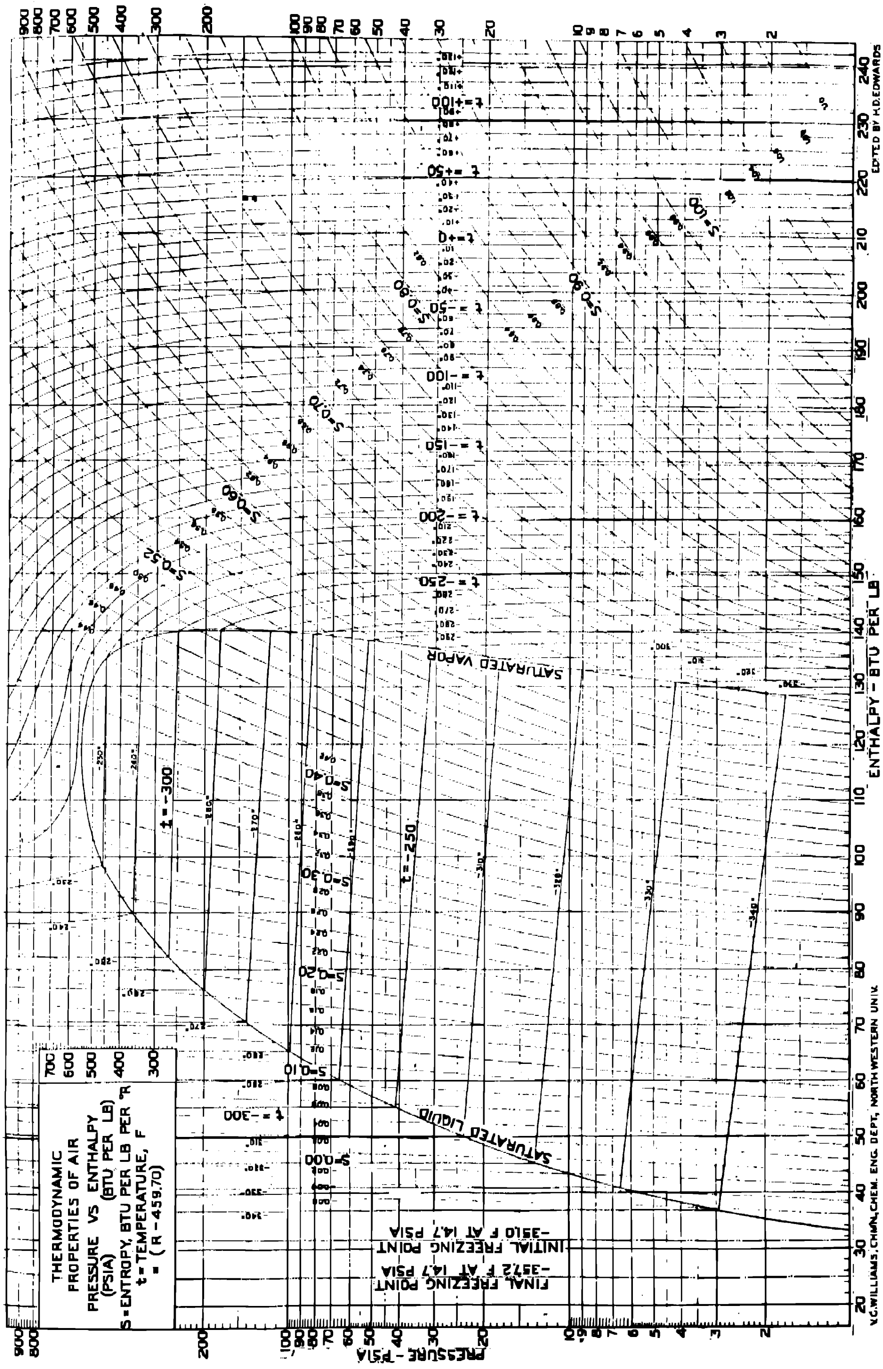
THERMODYNAMIC PROPERTIES OF AIR
PRESSURE VS ENTHALPY
(PSIA) (BTU PER LB)
S=ENTROPY, BTU PER LB PER °R
t=TEMPERATURE, F
=(R-459.70)
PRESSURE - PSIA
ENTHALPY - BTU PER LB
SATURATED VAPOR
SATURATED LIQUID
INITIAL FREEZING POINT
-351.0 F AT 14.7 PSIA
FINAL FREEZING POINT
-357.2 F AT 14.7 PSIA
t=+100
t=+50
t=0
t=-50
t=-100
t=-150
t=-200
t=-250
t=-300
t=-250
t=-300
S=1.00
S=0.90
S=0.80
S=0.70
S=0.60
S=0.52
S=0.40
S=0.30
S=0.20
S=0.10
S=0.00
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
10
9
8
7
6
5
4
3
2
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
210
220
230
240
V.C.WILLIAMS, CHEM. ENG. DEPT., NORTHWESTERN UNIV.
EDITED BY H.D.EDWARDS

Chart 1

Table 1. Ammonia—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−105	1.00	*27.9	45.71	223.14	−66.6	569.1	−0.1717	1.6211
−104	1.04	27.8	45.67	214.23	−65.6	569.6	− .1689	1.6174
−103	1.08	27.7	45.63	205.90	−64.5	570.1	− .1661	1.6136
−102	1.14	27.7	45.59	197.70	−63.5	570.6	− .1634	1.6100
−101	1.19	27.5	45.55	190.08	−62.5	571.0	− .1606	1.6062
−100	1.24	*27.4	45.51	182.90	−61.5	571.4	−0.1579	1.6025
− 99	1.29	27.3	45.47	175.42	−60.5	571.9	− .1552	1.5988
− 98	1.35	27.2	45.43	168.48	−59.5	572.3	− .1525	1.5952
− 97	1.41	27.0	45.40	161.98	−58.5	572.7	− .1498	1.5915
− 96	1.47	26.9	45.36	155.92	−57.5	573.2	− .1471	1.5878
− 95	1.52	*26.8	45.32	150.30	−56.5	573.6	−0.1444	1.5842
− 94	1.59	26.7	45.28	144.68	−55.5	574.1	− .1417	1.5806
− 93	1.66	26.5	45.24	139.27	−54.5	574.5	− .1390	1.5771
− 92	1.73	26.4	45.20	134.06	−53.4	575.0	− .1363	1.5736
− 91	1.79	26.2	45.16	129.06	−52.4	575.5	− .1336	1.5702
− 90	1.86	*26.1	45.12	124.28	−51.4	575.9	−0.1309	1.5667
− 89	1.94	26.0	45.08	119.75	−50.4	576.3	− .1281	1.5634
− 88	2.02	25.8	45.04	115.37	−49.4	576.7	− .1254	1.5600
− 87	2.11	25.6	45.00	111.31	−48.4	577.2	− .1227	1.5566
− 86	2.18	25.5	44.96	107.39	−47.4	577.6	− .1199	1.5533
− 85	2.27	*25.3	44.92	103.63	−46.4	578.0	−0.1171	1.5499
− 84	2.36	25.1	44.88	99.87	−45.4	578.4	− .1144	1.5466
− 83	2.46	24.9	44.84	96.28	−44.3	578.9	− .1117	1.5433
− 82	2.55	24.7	44.80	92.86	−43.3	579.3	− .1090	1.5401
− 81	2.65	24.5	44.76	89.65	−42.3	579.7	− .1063	1.5368
− 80	2.74	*24.3	44.73	86.54	−41.3	580.1	−0.1036	1.5336
− 79	2.85	24.1	44.68	83.50	−40.3	580.5	− .1009	1.5304
− 78	2.96	23.9	44.64	80.61	−39.2	581.0	− .0983	1.5273
− 77	3.07	23.6	44.60	77.90	−38.2	581.4	− .0956	1.5241
− 76	3.19	23.4	44.56	75.30	−37.2	581.8	− .0930	1.5210
− 75	3.30	*23.2	44.52	72.80	−36.2	582.2	−0.0903	1.5178
− 74	3.43	22.9	44.48	70.35	−35.2	582.7	− .0877	1.5148
− 73	3.56	22.7	44.44	68.01	−34.2	583.1	− .0800	1.5117
− 72	3.69	22.4	44.40	65.78	−33.1	583.6	− .0824	1.5087
− 71	3.82	22.2	44.36	63.70	−32.1	584.0	− .0797	1.5056
− 70	3.94	*21.9	44.32	61.65	−31.1	584.4	−0.0771	1.5026
− 69	4.09	21.6	44.28	59.60	−30.1	584.8	− .0745	1.4997
− 68	4.24	21.3	44.24	57.64	−29.4	585.3	− .0719	1.4969
− 67	4.39	21.0	44.19	55.78	−28.0	585.8	− .0694	1.4940
− 66	4.54	20.7	44.15	54.01	−27.0	586.2	− .0668	1.4911
− 65	4.69	*20.4	44.11	52.34	−26.0	586.6	−0.0642	1.4833
− 64	4.86	20.1	44.07	50.79	−25.0	587.0	− .0616	1.4856
− 63	5.03	19.6	44.03	49.26	−23.9	587.5	− .0590	1.4829
− 62	5.20	19.3	43.99	47.74	−22.9	588.0	− .0565	1.4802
− 61	5.38	18.9	43.95	46.23	−21.9	588.4	− .0539	1.4774
− 60	5.55	*18.6	43.91	44.73	−20.9	588.8	−0.0514	1.4747
− 59	5.74	18.2	43.87	43.37	−19.8	589.3	− .0487	1.4720
− 58	5.93	17.8	43.83	42.05	−18.8	589.7	− .0460	1.4694
− 57	6.13	17.4	43.78	40.79	−17.8	590.2	− .0434	1.4667
− 56	6.33	17.0	43.74	39.56	−16.7	590.6	− .0407	1.4640
− 55	6.54	*16.6	43.70	38.38	−15.7	591.0	−0.0381	1.4614
− 54	6.75	16.2	43.66	37.24	−14.6	591.5	− .0355	1.4588
− 53	6.97	15.7	43.62	36.15	−13.6	591.9	− .0330	1.4563
− 52	7.20	15.3	43.58	35.09	−12.6	592.3	− .0305	1.4537
− 51	7.43	14.8	43.54	34.06	−11.5	592.8	− .0279	1.4513

* Inches of mercury below one standard atmosphere (29.02 in.).

rated Vapor

40 F	Entropy, datur Btu per l
or	Liquid s_f
1	−0.1717
6	− .1689
1	− .1661
6	− .1634
0	− .1606
.4	−0 1579
.9	− .1552
.3	− .1525
7	− .1498
2	.1471
.6	−0.1444
.1	− .1417
.5	− .1390
.0	− .1363
.5	− .1336
.9	−0.1309
.3	− .1281
.7	− .1254
2	− .1227
.6	− .1199
0	−0.1171
4	− .1144
.9	− .1117
.3	− .1090
7	− .1063
.1	−0.1036
.5	− .1009
.0	− .0983
.4	− .0956
.8	− .0930
.2	−0.0903
.7	− .0877
.1	− .0800
6	− .0824
.0	− .0797
.4	−0.0771
.8	− .0745
3	− 0719
.8	− .0694
.2	− .0668
.6	- 0.0642
7.0	− .0616
7.5	− .0590
3.0	− .0565
3.4	− .0539
3.8	−0.0514
3.3	− .0487
9.7	− .0460
0.2	− .0434
0.6	− .0407
1.0	−0.0381
1.5	− .0355
1.9	− .0330
2.3	− .0305
2.8	− .0279

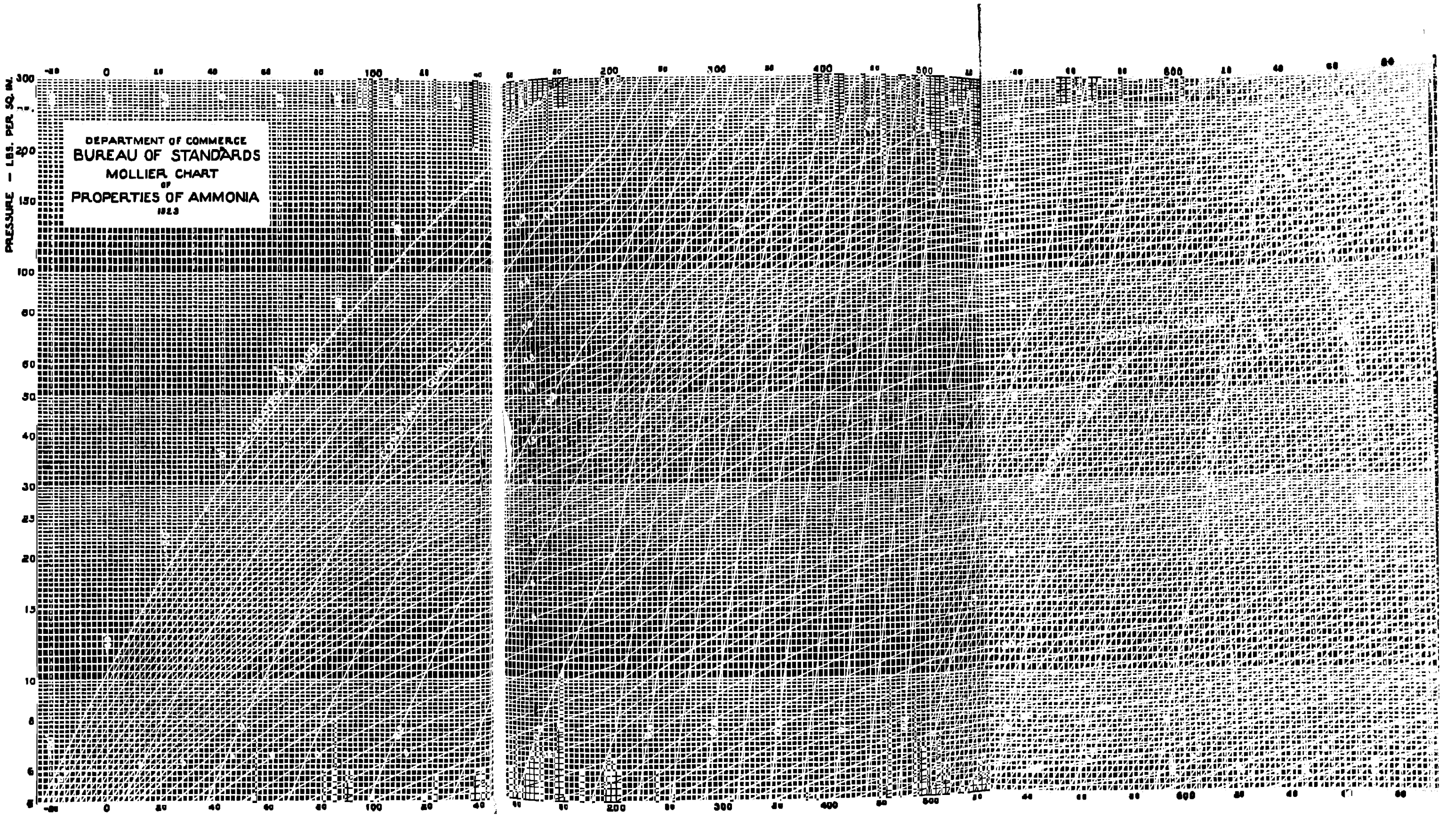

Chart 3. Ammonia (Bureau of Standards)

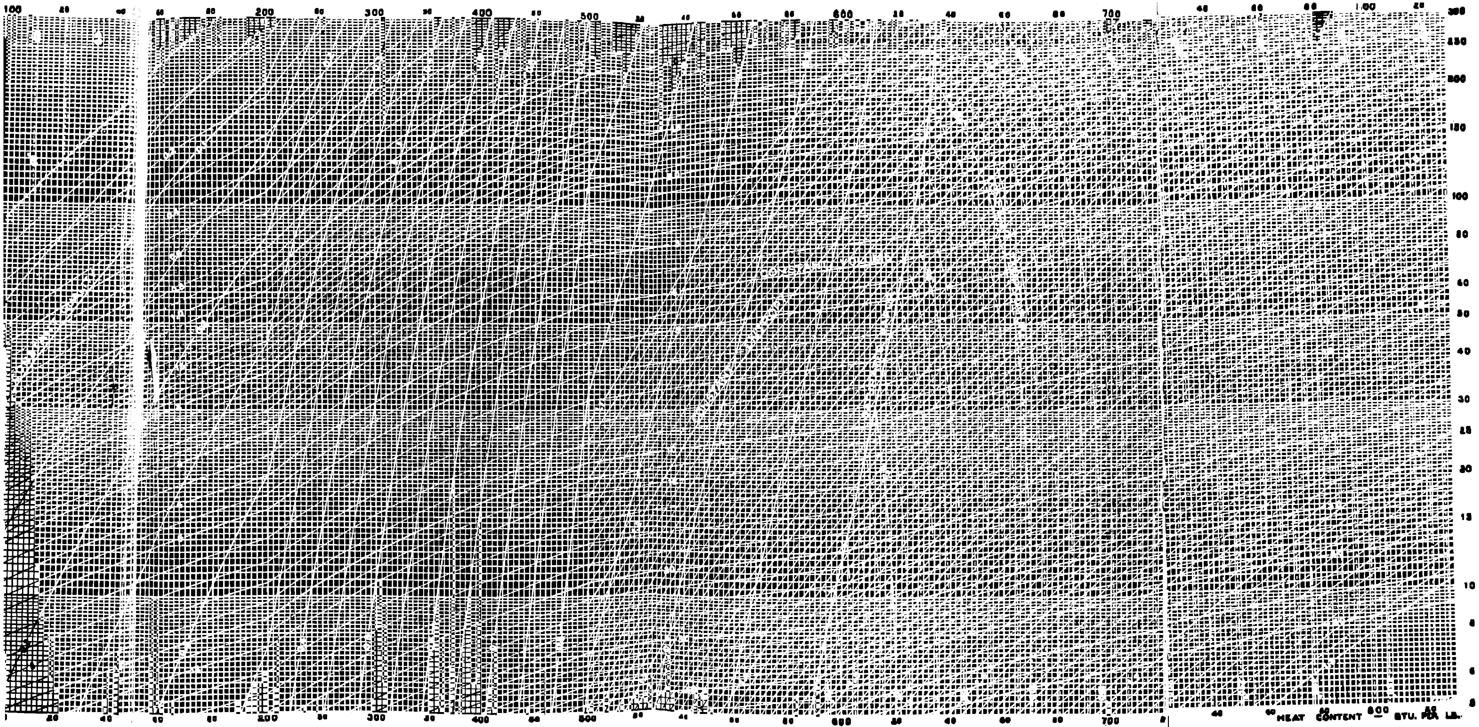

Chart 2. Ammonia (Bureau of Standards)

Table 1. Ammonia—Properties of Liquid and Saturated Vapor (Continued)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
− 50	7.67	*14.3	43.49	33.08	−10.5	593.2	−0.0254	1.4487
− 49	7.91	13.8	43.45	32.12	− 9.4	593.7	− .0229	1.4463
− 48	8.16	13.3	43.41	31.20	− 8.4	594.2	− .0203	1.4438
− 47	8.42	12.8	43.37	30.31	− 7.3	594.6	− .0178	1.4413
− 46	8.68	12.2	43.33	29.45	− 6.3	595.0	− .0153	1.4388
− 45	8.95	*11.7	43.28	28.62	− 5.3	595.4	−0.0128	1.4363
− 44	9.23	11.1	43.24	27.82	− 4.3	595.9	− .0102	1.4338
− 43	9.51	10.6	43.20	27.04	− 3.2	596.4	− .0076	1.4314
− 42	9.81	10.0	43.16	26.29	− 2.1	596.8	− .0051	1.4290
− 41	10.10	9.3	43.12	25.56	− 1.1	597.2	− .0025	1.4266
− 40	10.41	*8.7	43.08	24.86	0.0	597.6	0.0000	1.4242
− 39	10.72	8.1	43.04	24.18	1.1	598.0	.0025	1.4217
− 38	11.04	7.4	42.99	23.53	2.1	598.3	.0051	1.4193
− 37	11.37	6.8	42.95	22.89	3.2	598.7	.0076	1.4169
− 36	11.71	6.1	42.90	22.27	4.3	599.1	.0101	1.4144
− 35	12.05	*5.4	42.86	21.68	5.3	599.5	0.0126	1.4120
− 34	12.41	4.7	42.82	21.10	6.4	599.9	.0151	1.4096
− 33	12.77	3.9	42.78	20.54	7.4	600.2	.0176	1.4072
− 32	13.14	3.2	42.73	20.00	8.5	600.6	.0201	1.4048
− 31	13.52	2.4	42.69	19.48	9.6	601.0	.0226	1.4025
− 30	13.90	*1.6	42.65	18.97	10.7	601.4	0.0250	1.4001
− 29	14.30	0.8	42.61	18.48	11.7	601.7	.0275	1.3978
− 28	14.71	0.0	42.57	18.00	12.8	602.1	.0300	1.3955
− 27	15.12	0.4	42.54	17.54	13.9	602.5	.0325	1.3932
− 26	15.55	0.8	42.48	17.09	14.9	602.8	.0350	1.3909
− 25	15.98	1.3	42.44	16.66	16.0	603.2	0.0374	1.3886
− 25	16.42	1.7	42.40	16.24	17.1	603.6	.0399	1.3863
− 23	16.88	2.2	42.35	15.83	18.1	603.9	.0423	1.3840
− 22	17.34	2.6	42.31	15.43	19.2	604.3	.0488	1.3818
− 21	17.81	3.1	42.26	15.05	20.3	604.6	.0472	1.3796
− 20	18.30	3.6	42.22	14.68	21.4	605.0	0.0497	1.3774
− 19	18.79	4.1	42.18	14.32	22.4	605.3	.0521	1.3752
− 18	19.30	4.6	42.13	13.97	23.5	605.7	.0545	1.3729
− 17	19.81	5.1	42.09	13.62	24.6	606.1	.0570	1.3708
− 16	20.34	5.6	42.04	13.29	25.6	606.4	.0594	1.3686
− 15	20.88	6.2	42.00	12.97	26.7	606.7	0.0618	1.3664
− 14	21.43	6.7	41.96	12.66	27.8	607.1	.0642	1.3643
− 13	21.99	7.8	41.91	12.36	28.9	607.5	.0666	1.3621
− 12	22.56	7.9	41.87	12.06	30.0	607.8	.0690	1.3600
− 11	23.15	8.5	41.82	11.78	31.0	608.1	.0714	1.3579
− 10	23.74	9.0	41.78	11.50	32.1	608.5	0.0738	1.3558
− 9	24.35	9.7	41.74	11.23	33.2	608.8	.0762	1.3537
− 8	24.97	10.3	41.69	10.97	34.3	609.2	.0768	1.3516
− 7	25.61	10.9	41.65	10.71	35.4	609.5	.0809	1.3495
− 6	26.26	11.6	41.60	10.47	36.4	609.8	.0833	1.3474
− 5	26.92	12.2	41.56	10.23	37.5	610.1	0.0857	1.3454
− 4	27.59	12.9	41.52	9.991	38.6	610.5	.0880	1.3433
− 3	28.28	13.6	41.47	9.763	39.7	610.8	.0904	1.3413
− 2	28.98	14.3	41.43	9.541	40.7	611.1	.0928	1.3393
− 1	29.69	15.0	41.38	9.326	14.8	611.4	.0951	1.3372
0	30.42	15.7	41.34	9.116	42.9	611.8	0.0975	1.3352
1	31.16	16.5	41.29	8.912	44.0	612.1	0.0998	1.3332
2	31.92	17.2	41.25	8.714	45.1	612.4	.1022	1.3312
3	32.69	18.0	41.20	8.521	46.2	612.7	.1045	1.3292
4	33.47	18.8	41.16	8.333	47.2	613.0	.1069	1.3273
5	34.27	19.6	41.11	8.150	48.3	613.3	.1092	1.3253
6	35.09	20.4	41.07	7.971	49.4	613.6	0.1115	1.3234
7	35.92	21.2	41.01	7.798	50.5	613.9	.1138	1.3214
8	36.77	22.1	40.98	8.629	51.6	614.3	.1162	1.3195
9	37.63	22.9	40.93	7.464	52.7	614.6	.1185	1.3176
10	38.51	23.8	40.89	7.304	53.8	614.9	.1208	1.3157
11	39.40	24.7	40.84	7.148	54.9	615.2	0.1231	1.3137
12	40.31	25.6	40.80	6.996	56.0	615.5	.1254	1.3118
13	41.24	26.5	40.75	6.847	57.1	615.8	.1277	1.3099
14	42.18	27.5	40.71	6.703	58.2	616.1	.1300	1.3081
15	43.14	28.4	40.66	6.562	59.2	616.3	.1323	1.3062

* Inches of mercury below one standard atmosphere (29.92 in.).

Table 1. Ammonia—Properties of Liquid and Saturated Vapor (Continued)

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
16	44.12	29.4	40.61	6.425	60.3	616.6	0.1346	1.3043
17	45.12	30.4	40.57	6.291	61.4	616.9	.1369	1.3025
18	46.13	31.4	40.52	6.161	62.5	617.2	.1392	1.3006
19	47.16	32.5	40.48	6.034	63.6	617.5	.1415	1.2988
20	48.21	33.5	40.43	5.910	64.7	617.8	.1437	1.2969
21	49.28	34.6	40.38	5.789	65.8	618.0	0.1460	1.2951
22	50.36	35.7	40.34	5.671	66.9	618.3	.1483	1.2933
23	51.47	36.8	40.29	5.556	68.0	618.6	.1505	1.2951
24	52.59	37.9	40.25	5.443	69.1	618.9	.1528	1.2897
25	53.78	39.0	40.20	5.334	70.2	619.1	.1551	1.2879
26	54.90	40.2	40.15	5.227	71.3	619.4	0.1573	1.2861
27	56.08	41.4	40.10	5.123	72.4	619.7	.1596	1.2843
28	57.28	42.6	40.06	5.021	73.5	619.9	.1618	1.2825
29	58.50	43.8	40.01	4.922	74.6	620.2	.1641	1.2808
30	59.74	45.0	39.96	4.825	75.7	620.5	.1663	1.2790
31	61.00	46.3	39.91	4.730	76.8	620.7	0.1686	1.2773
32	62.29	47.6	39.86	4.637	77.9	621.0	.1708	1.2755
33	63.59	48.9	39.82	4.547	79.0	621.2	.1730	1.2738
34	64.91	50.2	39.77	4.459	80.1	621.5	.1753	1.2731
35	66.26	52.6	39.72	4.373	81.2	621.7	.1775	1.2704
36	67.63	52.9	39.67	4.289	82.3	622.0	0.1797	1.2686
37	69.02	54.3	39.63	4.207	83.4	622.2	.1819	1.2669
38	70.43	55.7	39.58	4.126	84.6	622.5	.1841	1.2652
39	71.87	57.2	39.54	4.048	85.7	622.7	.1863	1.2635
40	73.32	58.6	39.49	3.971	86.8	623.0	.1885	1.2618
41	74.80	60.1	39.44	3.897	87.9	623.2	0.1908	1.2602
42	76.31	61.6	39.39	3.823	89.0	623.4	.1930	1.2585
43	77.83	63.1	39.34	3.752	90.1	623.7	.1952	1.2568
44	79.38	64.7	39.29	3.682	91.2	623.9	.1974	1.2552
45	80.96	66.3	39.24	3.614	92.3	624.1	.1996	1.2535
46	82.55	67.9	39.19	3.547	93.5	624.4	0.2018	1.2519
47	84.18	69.5	39.14	3.481	94.6	624.6	.2040	1.2502
48	85.82	71.1	39.10	3.418	95.7	624.8	.2062	1.2486
49	87.49	72.8	39.05	3.355	96.8	625.0	.2083	1.2469
50	89.19	74.5	39.00	3.294	97.9	625.2	.2105	1.2453
51	90.91	76.2	38.95	3.234	99.1	625.5	0.2127	1.2437
52	92.66	78.0	38.90	3.176	100.2	625.7	.2149	1.2421
53	94.43	79.7	38.85	3.119	101.3	625.9	..2171	1.2405
54	96.23	81.5	38.80	3.063	102.4	626.1	.2192	1.2389
55	98.06	83.4	38.75	3.008	103.5	626.3	.3214	1.2373
56	99.91	85.2	38.70	2.954	104.7	626.5	0.2236	1.2357
57	101.8	87.1	38.65	2.902	105.8	626.7	.2257	1.2341
58	103.7	89.0	38.60	2.851	106.9	626.9	.2279	1.2325
59	105.6	90.9	38.55	2.800	108.1	627.1	.2301	1.2310
60	107.6	92.9	38.50	2.751	109.2	627.3	.2322	1.2294
61	109.6	94.9	38.45	2.703	110.3	627.5	0.2344	1.2278
62	111.6	96.9	38.40	2.656	111.5	627.7	.2365	1.2262
63	113.6	98.9	38.35	2.610	112.6	627.9	.2387	1.2247
64	115.7	101.0	38.30	2.565	113.7	628.0	.2408	1.2231
65	117.8	103.1	38.25	2.520	114.8	628.2	.2430	1.2216
66	120.0	105.3	38.20	2.477	116.0	628.4	0.2451	1.2201
67	122.1	107.4	38.15	2.435	117.1	628.6	.2473	1.2186
68	124.3	109.6	38.10	2.393	118.3	628.8	.2494	1.2170
69	126.5	111.8	38.05	2.352	119.4	628.9	.2515	1.2155
70	128.8	114.1	38.00	2.312	120.5	629.1	.2537	1.2140
71	131.1	116.4	37.95	2.273	121.7	629.3	0.2558	1.2125
72	133.4	118.7	37.90	2.235	122.8	629.4	.2579	1.2110
73	135.7	121.0	37.84	2.197	124.0	629.6	.2601	1.2095
74	138.1	123.4	37.79	2.161	125.1	629.8	.2622	1.2080
75	140.5	125.8	37.74	2.125	126.2	629.9	.2643	1.2065
76	143.0	126.3	37.69	2.089	127.4	630.1	0.2664	1.2050
77	145.4	130.7	37.64	2.055	128.5	630.2	.2685	1.2035
78	147.9	133.2	37.58	2.021	129.7	630.4	.2706	1.2020
79	150.5	135.8	37.53	1.988	130.8	630.5	.2728	1.2006
80	153.0	138.3	37.48	1.955	132.0	630.7	.2749	1.1991

Table 1. Ammonia—Properties of Liquid and Saturated Vapor (Concluded)

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
81	155.6	140.9	37.43	1.923	133.1	630.8	0.2769	1.1976
82	158.3	143.6	37.37	1.892	134.3	631.0	.2791	1.1962
83	161.0	146.3	37.32	1.861	135.4	631.1	.2812	1.1947
84	163.6	149.0	37.26	1.831	136.6	631.3	.2833	1.1933
85	166.4	151.7	37.21	1.801	137.8	631.4	.2854	1.1918
86	169.2	154.5	37.16	1.772	138.9	631.5	0.2875	1.1904
87	172.0	157.3	37.11	1.744	140.1	631.7	.2895	1.1889
88	174.8	160.1	37.05	1.716	141.2	631.8	.2917	1.1875
89	177.7	163.0	37.00	1.688	142.4	631.9	.2937	1.1860
90	180.6	165.9	36.95	1.661	143.5	632.0	.2958	1.1846
91	183.6	168.9	36.89	1.635	144.7	632.1	0.2979	1.1832
92	186.6	171.9	36.84	1.609	145.8	632.3	.3000	1.1818
93	189.6	174.9	36.78	1.584	147.0	632.3	.3021	1.1804
94	192.7	178.0	36.73	1.559	148.2	632.5	.3041	1.1789
95	195.8	181.1	36.67	1.534	149.4	632.6	.3062	1.1775
96	198.9	184.2	36.62	1.510	150.5	632.6	0.3083	1.1761
97	202.1	187.4	36.56	1.487	151.7	632.8	.3104	1.1747
98	205.3	190.6	36.51	1.464	152.9	632.9	.3125	1.1733
99	208.6	193.9	36.45	1.441	154.0	632.9	.3145	1.1719
100	211.9	197.2	36.40	1.419	155.2	633.0	.3166	1.1705
101	215.2	200.5	36.34	1.397	156.4	633.1	0.3187	1.1691
102	218.6	203.9	36.29	1.375	157.6	633.2	.3207	1.1677
103	222.0	207.3	36.23	1.354	158.7	633.3	.3228	1.1663
104	224.4	210.7	36.18	1.334	159.9	633.4	.3248	1.1649
105	228.9	214.2	36.12	1.313	161.1	633.4	.3269	1.1635
106	232.5	217.8	36.06	1.293	162.3	633.5	0.3289	1.1621
107	236.0	221.3	36.01	1.274	163.5	633.6	.3310	1.1607
108	239.7	225.0	35.95	1.254	164.6	633.6	.3330	1.1593
109	243.3	228.6	35.90	1.235	165.8	633.7	.3351	1.1580
110	247.0	232.3	35.84	1.217	167.0	633.7	.3372	1.1566
111	250.8	236.1	35.78	1.198	168.2	633.8	0.3392	1.1552
112	354.5	239.8	35.72	1.180	169.4	633.8	.3413	1.1538
113	258.4	243.7	35.67	1.163	170.6	633.9	.3433	1.1524
114	262.2	247.5	35.61	1.145	171.8	633.9	.3453	1.1510
115	266.2	251.5	35.55	1.128	173.0	633.9	.3474	1.1497
116	270.1	255.4	35.49	1.112	174.2	634.0	0.3495	1.1483
118	274.1	259.4	35.43	1.095	175.4	634.0	.3515	1.1469
118	278.2	263.5	35.38	1.079	176.6	634.0	.3515	1.1455
119	282.3	267.6	35.32	1.063	177.8	634.0	.3556	1.1441
120	286.4	271.7	35.26	1.047	179.0	634.0	.3576	1.1427
121	290.6	275.9	35.20	1.032	180.2	634.0	0.3597	1.1414
122	294.8	280.1	35.14	1.017	181.4	634.0	.3618	1.1400
123	299.1	284.4	35.08	1.002	182.6	634.0	.3638	1.1386
124	303.4	288.7	35.02	0.987	183.9	634.0	.3659	1.1372
125	307.8	293.1	34.96	0.973	185.1	634.0	.3679	1.1358

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb)

Temp, F	Pressure in psia									Saturation temperature in italics, F				
	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	−63.11	*−57.64*	*−52.88*	*−48.64*	*−44.38*	*−41.34*	*−38.14*	*−35.16*	*−32.37*	*−29.76*	*−27.29*	*−24.95*	*−22.73*	*−20 61*
(at sat)	*(49.31)*	*(41.59)*	*(36.01)*	*(31.79)*	*(28.48)*	*(25.81)*	*(23.61)*	*(21.77)*	*(20.20)*	*(18.86)*	*(17.67)*	*(16.64)*	*(15.72)*	*(14.90)*
−50	51.05	42.44	36.29	—	—	—	—	—	—	—	—	—	—	—
−40	52.36	43.55	37.25	32.52	28.85	25.90	—	—	—	—	—	—	—	—
−30	53.67	44.64	38.19	33.36	29.59	26.58	24.12	22.07	20.33	—	—	—	—	—
−20	54.97	45.73	39.13	34.19	30.34	27.26	24.74	22.64	20.86	19.33	18.01	16.86	15.83	14.93
−10	56.26	46.82	40.07	35.01	31.07	27.92	25.35	23.20	21.38	19.82	18.47	17.29	16.24	15.32
0	57.55	47.90	41.00	35.83	31.80	28.58	25.95	23.75	21.90	20.30	18.92	17.72	16.65	15.70
10	58.84	48.98	41.93	36.64	32.53	29.24	26.55	24.31	22.41	20.78	19.37	18.14	17.05	16.08
20	60.12	50.05	42.85	37.45	33.26	29.90	27.15	24.86	22.92	21.26	19.82	18.56	17.45	16.46
30	61.41	51.12	43.77	38.26	33.98	30.55	27.74	25.41	23.43	21.73	20.26	18.97	17.84	16.83
40	62.69	52.19	44.69	39.07	34.70	31.20	28.34	25.95	23.93	22.20	20.70	19.39	18.23	17.20
50	63.96	53.26	45.61	39.88	35.42	31.85	28.93	26.49	24.43	22.67	21.14	19.80	18.62	17.57
60	65.24	54.32	46.53	40.68	36.13	32.49	29.52	27.03	24.94	23.14	21.58	20.21	19.01	17.94
70	66.51	55.39	47.44	41.48	36.85	33.14	30.10	27.57	25.43	23.60	22.01	20.62	19.39	18.30
80	67.79	56.45	48.36	42.28	37.56	33.78	30.69	28.11	25.93	24.06	22.44	21.03	19.78	18.67
90	69.06	57.51	49.27	43.08	38.27	34.42	31.28	28.65	26.43	24.53	22.88	21.43	20.16	19.03
100	70.33	58.58	50.18	43.88	38.98	35.07	31.86	29.19	26.93	24.99	23.31	21.84	20.54	19.39
110	71.60	59.64	51.09	44.68	39.70	35.71	32.44	29.72	27.42	25.45	23.74	22.24	20.92	19.75
120	72.87	60.70	52.00	45.48	40.40	36.35	33.03	30.26	27.92	25.91	24.17	22.65	21.30	20.11
130	74.14	61.76	52.91	46.27	41.11	36.99	33.61	30.79	28.41	26.37	24.60	23.05	21.68	20.47
140	75.41	62.82	53.82	47.07	41.82	37.62	34.19	31.33	28.90	26.85	25.03	23.45	22.06	20.83
150	76.68	63.87	54.73	47.87	42.53	38.26	34.77	31.86	29.40	27.29	25.46	23.86	22.44	21.19
160	77.95	64.93	55.63	48.66	43.24	38.90	35.35	32.39	29.89	27.74	25.88	24.26	22.82	21.54
170	79.21	65.99	56.54	49.46	43.95	39.54	35.93	32.92	30.38	28.20	26.31	24.66	23.20	21.90
180	80.48	67.05	57.45	50.25	44.65	40.17	36.51	33.46	30.87	28.66	26.74	25.06	23.58	22.26
190	—	—	—	—	—	40.81	37.09	33.99	31.36	29.11	27.16	25.46	23.95	22.61
200	—	—	—	—	—	41.45	37.67	34.52	31.85	29.57	27.59	25.86	24.33	22.97
220	—	—	—	—	—	—	—	—	—	—	28.44	26.66	25.08	23.68

Temp, F	Pressure in psia									Saturation temperature in italics, F				
	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	−20.61	*−18.58*	*−16.64*	*−14.78*	*−12.98*	*−11.25*	*−9.58*	*−7.96*	*−6.39*	*−4.87*	*−3.40*	*−1.97*	*−0.57*	*+0.79*
(at sat)	*(14.90)*	*(14.17)*	*(13.50)*	*(12.90)*	*(12.35)*	*(11.85)*	*(11.39)*	*(10.98)*	*(10.56)*	*(10.20)*	*(9.853)*	*(9.534)*	*(9.236)*	*(8.955)*
−20	14.93	—	—	—	—	—	—	—	—	—	—	—	—	—
−10	15.32	14.49	13.74	13.06	12.45	11.89	—	—	—	—	—	—	—	—
0	15.70	14.85	14.09	13.40	12.77	12.20	11.67	11.19	10.74	10.33	9.942	9.584	9.250	—
10	16.08	15.21	14.44	13.73	13.09	12.50	11.96	11.47	11.01	10.59	10.20	9.834	9.492	9.173
20	16.46	15.57	14.78	14.06	13.40	12.80	12.25	11.75	11.28	10.85	10.45	10.08	9.731	9.405
30	16.83	15.93	15.11	14.38	13.71	13.10	12.54	12.03	11.55	11.11	10.70	10.32	9.966	9.633
40	17.20	16.28	15.45	14.70	14.02	13.40	12.82	12.30	11.81	11.37	10.95	10.56	10.20	9.858
50	17.57	16.63	15.78	15.02	14.32	13.69	13.11	12.57	12.08	11.62	11.19	10.80	10.43	10.08
60	17.94	16.98	16.12	15.34	14.63	13.98	13.39	12.84	12.34	11.87	11.44	11.03	10.65	10.30
70	18.30	17.33	16.45	15.65	14.93	14.27	13.66	13.11	12.59	12.12	11.68	11.26	10.88	10.52
80	18.67	17.67	16.78	15.97	15.23	14.56	13.94	13.37	12.85	12.37	11.92	11.50	11.10	10.74
90	19.03	18.02	17.10	16.28	15.53	14.84	14.22	13.64	13.11	12.61	12.15	11.73	11.33	10.96
100	19.39	18.36	17.43	16.59	15.83	15.13	14.49	13.90	13.36	12.86	12.39	11.96	11.55	11.17
110	19.75	18.70	17.76	16.90	16.12	15.41	14.76	14.17	13.61	13.10	12.63	12.18	11.77	11.38
120	20.11	19.04	18.08	17.21	16.42	15.70	15.04	14.43	13.87	13.34	12.86	12.41	11.99	11.60
130	20.47	19.38	18.41	17.52	16.72	15.98	15.31	14.69	14.12	13.59	13.10	12.64	12.21	11.81
140	20.83	19.72	18.73	17.83	17.01	16.26	15.58	14.95	14.37	13.83	13.33	12.86	12.43	12.02
150	21.19	20.06	19.05	18.14	17.31	16.55	15.85	15.21	14.62	14.07	13.56	13.09	12.65	12.23
160	21.54	20.40	19.37	18.44	17.60	16.83	16.12	15.47	14.87	14.31	13.80	13.31	12.87	12.44
170	21.90	20.74	19.70	18.75	17.89	17.11	16.39	15.73	15.12	14.55	14.03	13.54	13.08	12.66
180	22.26	21.08	20.02	19.06	18.19	17.39	16.66	15.99	15.37	14.79	14.26	13.76	13.30	12.87
190	22.61	21.42	20.34	19.36	18.48	17.67	16.93	16.25	15.62	15.03	14.49	13.99	13.52	13.07
200	22.97	21.75	20.66	19.67	18.77	17.95	17.20	16.50	15.86	15.27	14.72	14.21	13.73	13.28
220	23.68	22.43	21.30	20.28	19.35	18.51	17.73	17.02	16.36	15.75	15.18	14.65	14.16	13.70
240	—	—	21.94	20.89	19.94	19.07	18.27	17.53	16.85	16.23	15.64	15.10	14.59	14.12
260	—	—	—	—	—	—	—	18.04	17.35	16.70	16.10	15.54	15.02	14.53
280	—	—	—	—	—	—	—	—	—	—	—	—	15.45	14.95

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb) (Continued)

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	31	32	33	34	35	36	37	38	39	40	42	44	46	48
	0.79	*2.11*	*3.40*	*4.66*	*5.89*	*7.09*	*8.27*	*9.42*	*10.55*	*11.66*	*13.81*	*15.88*	*17.87*	*19.80*
(at sat)	*(8.955)*	*(8.693)*	*(8.445)*	*(8.211)*	*(7.991)*	*(7.782)*	*(7.584)*	*(7.396)*	*(7.217)*	*(7.047)*	*(6.731)*	*(6.442)*	*(6.177)*	*(5.934)*
10	9.173	8.874	8.592	8.328	8.078	7.842	7.619	7.407	—	—	—	—	—	—
20	9.405	9.099	8.812	8.542	8.287	8.046	7.819	7.603	7.398	7.203	6.842	6.513	6.213	5.937
30	9.633	9.321	9.028	8.753	8.493	8.247	8.015	7.795	7.586	7.387	7.019	6.683	6.377	6.096
40	9.858	9.540	9.242	8.960	8.695	8.445	8.208	7.983	7.770	7.568	7.192	6.850	6.538	6.251
50	10.08	9.757	9.452	9.166	8.895	8.640	8.398	8.170	7.952	7.746	7.363	7.014	6.696	6.404
60	10.30	9.972	9.661	9.369	9.093	8.833	8.587	8.353	8.132	7.922	7.531	7.176	6.851	6.554
70	10.52	10.18	9.868	9.570	9.289	9.024	8.773	8.535	8.310	8.096	7.697	7.336	7.005	6.702
80	10.74	10.40	10.07	9.770	9.484	9.214	8.958	8.716	8.486	8.268	7.862	7.494	7.157	6.848
90	10.96	10.61	10.28	9.969	9.677	9.402	9.142	8.895	8.661	8.439	8.026	7.650	7.308	6.993
100	11.17	10.81	10.48	10.17	9.869	9.589	9.324	9.073	8.835	8.609	8.188	7.806	7.457	7.137
110	11.38	11.02	10.68	10.36	10.06	9.775	9.506	9.250	9.008	8.777	8.349	7.960	7.605	7.280
120	11.60	11.23	10.88	10.56	10.25	9.961	9.686	9.426	9.179	8.945	8.510	8.114	7.753	7.421
130	11.81	11.44	11.08	10.75	10.44	10.15	9.866	9.602	9.351	9.112	8.669	8.267	7.899	7.562
140	12.02	11.64	11.28	10.95	10.63	10.33	10.05	9.776	9.521	9.278	8.828	8.419	8.045	7.702
150	12.23	11.85	11.48	11.14	10.82	10.51	10.22	9.950	9.691	9.444	8.986	8.570	8.190	7.842
160	12.44	12.05	11.68	11.33	11.00	10.69	10.40	10.12	9.860	9.609	9.144	8.721	8.335	7.981
170	12.66	12.26	11.88	11.53	11.19	10.88	10.58	10.30	10.03	9.774	9.301	8.871	8.479	8.119
180	12.87	12.46	12.08	11.72	11.38	11.06	10.76	10.47	10.20	9.938	9.458	9.021	8.623	8.257
190	13.07	12.66	12.27	11.91	11.56	11.24	10.93	10.64	10.36	10.10	9.614	9.171	8.766	8.395
200	13.28	12.86	12.47	12.10	11.75	11.42	11.11	10.81	10.53	10.27	9.770	9.320	8.909	8.532
220	13.70	13.27	12.86	12.48	12.12	11.78	11.46	11.16	10.87	10.59	10.08	9.617	9.194	8.805
240	14.12	13.67	13.26	12.86	12.49	12.14	11.81	11.50	11.20	10.92	10.39	9.913	9.477	9.077
260	14.53	14.08	13.65	13.24	12.86	12.50	12.16	11.84	11.53	11.24	10.70	10.21	9.760	9.348
280	14.95	14.48	14.04	13.62	13.23	12.86	12.51	12.18	11.86	11.56	11.01	10.50	10.04	9.619
300	—	—	—	—	—	—	—	—	—	11.88	11.31	10.80	10.32	9.888

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	48	50	52	54	56	58	60	62	64	66	68	70	75	80
	19.80	*21.67*	*23.48*	*25.23*	*26.94*	*28.59*	*30.21*	*31.78*	*33.31*	*34.81*	*36.27*	*37.70*	*41.13*	*44.40*
(at sat)	*(5.934)*	*(5.710)*	*(5.502)*	*(5.309)*	*(5.129)*	*(4.962)*	*(4.805)*	*(4.658)*	*(4.519)*	*(4.389)*	*(4.267)*	*(4.151)*	*(3.887)*	*(3.655)*
20	5.937	—	—	—	—	—	—	—	—	—	—	—	—	—
30	6.096	5.838	5.599	5.378	5.172	4.981	—	—	—	—	—	—	—	—
40	6.251	5.988	5.744	5.519	5.310	5.115	4.933	4.762	4.602	4.452	4.310	4.177	—	—
50	6.404	6.135	5.887	5.657	5.444	5.245	5.060	4.886	4.723	4.570	4.426	4.290	3.982	3.712
60	6.554	6.280	6.027	5.793	5.576	5.373	5.184	5.007	4.842	4.686	4.539	4.401	4.087	3.812
70	6.702	6.423	6.165	5.927	5.706	5.499	5.307	5.127	4.958	4.799	4.650	4.509	4.189	3.909
80	6.848	6.564	6.302	6.059	5.834	5.624	5.428	5.244	5.072	4.910	4.758	4.615	4.289	4.005
90	6.993	6.704	6.437	6.190	5.960	5.746	5.547	5.360	5.185	5.020	4.865	4.719	4.388	4.098
100	7.137	6.843	6.571	6.319	6.085	5.868	5.665	5.474	5.296	4.129	4.971	4.822	4.485	4.190
110	7.280	6.980	6.704	6.447	6.209	5.988	5.781	5.588	5.406	5.236	5.075	4.924	4.581	4.281
120	7.421	7.117	6.835	6.575	6.333	6.107	5.897	5.700	5.516	5.342	5.179	5.025	4.676	4.371
130	7.562	7.252	6.966	6.701	6.455	6.226	6.012	5.811	5.624	5.447	5.281	5.125	4.770	4.460
140	7.702	7.387	7.096	6.827	6.576	6.343	6.126	5.922	5.731	5.552	5.383	5.224	4.863	4.548
150	7.842	7.521	7.225	6.952	6.697	6.460	6.239	6.032	5.838	5.656	5.484	5.323	4.956	4.635
160	7.981	7.655	7.354	7.076	6.817	6.577	6.352	6.142	5.944	5.759	5.585	5.420	5.048	4.722
170	8.119	7.788	7.483	7.200	6.937	6.692	6.464	6.250	6.050	5.862	5.685	5.518	5.139	4.808
180	8.257	7.921	7.611	7.323	7.056	6.808	6.576	6.359	6.155	5.964	5.784	5.615	5.230	4.893
190	8.395	8.053	7.738	7.446	7.175	6.923	6.687	6.467	6.260	6.066	5.883	5.711	5.320	4.978
200	8.532	8.185	7.865	7.569	7.294	7.037	6.787	6.574	6.364	6.167	5.982	5.807	5.410	5.063
210	8.669	8.317	7.992	7.691	7.412	7.151	6.909	6.681	6.468	6.268	6.080	5.902	5.500	5.147
220	8.805	8.448	8.118	7.813	7.529	7.265	7.019	6.788	6.572	6.369	6.179	5.998	5.589	5.231
230	8.941	8.579	8.244	7.935	7.647	7.379	7.129	6.895	6.675	6.470	6.275	6.093	5.678	5.315
240	9.077	8.710	8.370	8.056	7.764	7.492	7.238	7.001	6.778	6.570	6.373	6.187	5.767	5.398
250	9.213	8.840	8.496	8.177	7.881	7.605	7.338	7.107	6.881	6.670	6.470	6.282	5.855	5.482
260	9.348	8.970	8.621	8.298	7.998	7.718	7.437	7.213	6.984	6.769	6.567	6.376	5.943	5.565
280	9.619	9.230	8.871	8.539	8.230	7.943	7.676	7.424	7.188	6.968	6.760	6.563	6.119	5.730
300	9.888	9.489	9.120	8.779	8.462	8.167	7.892	7.634	7.392	7.165	6.952	6.750	6.294	5.894

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb) (Continued)

Temp, F	Pressure in psia							Saturation temperature in italics, F						
	80	85	90	95	100	105	110	115	120	125	130	135	140	145
	44.40	*47.60*	*50.47*	*53.32*	*56.05*	*58.67*	*61.21*	*63.66*	*66.02*	*68.31*	*70.53*	*72.69*	*74.79*	*76.83*
(at sat)	*(3.655)*	*(3.449)*	*(3.266)*	*(3.101)*	*(2.952)*	*(2.817)*	*(2.693)*	*(2.580)*	*(2.476)*	*(2.380)*	*(2.291)*	*(2.209)*	*(2.132)*	*(2.061)*
50	3.712	3.473	—	—	—	—	—	—	—	—	—	—	—	—
60	3.812	3.569	3.353	3.160	2.985	2.828	—	—	—	—	—	—	—	—
70	3.909	3.662	3.442	3.245	3.068	2.907	2.761	2.628	2.505	2.392	—	—	—	—
80	4.005	3.753	3.529	3.329	3.149	2.985	2.837	2.701	2.576	2.461	2.355	2.257	2.166	2.080
90	4.098	3.842	3.614	3.411	3.227	3.061	2.910	2.772	2.645	2.528	2.421	2.321	2.228	2.141
100	4.190	3.930	3.698	3.491	3.304	3.135	2.981	2.841	2.712	2.593	2.484	2.382	2.288	2.200
110	4.281	4.016	3.780	3.570	3.380	3.208	3.051	2.909	2.778	2.657	2.546	2.442	2.347	2.257
120	4.371	4.101	3.862	3.647	3.454	3.279	3.120	2.975	2.842	2.719	2.606	2.501	2.404	2.313
130	4.460	4.186	3.942	3.724	3.527	3.350	3.188	3.040	2.905	2.780	2.665	2.559	2.460	2.368
140	4.548	4.269	4.021	3.799	3.600	3.419	3.255	3.105	2.967	2.840	2.724	2.615	2.515	2.421
150	4.635	4.352	4.100	3.874	3.672	3.488	3.321	3.168	3.029	2.900	2.781	2.671	2.569	2.474
160	4.722	4.434	4.178	3.949	3.743	3.556	3.386	3.231	3.089	2.958	2.838	2.726	2.622	2.526
170	4.808	4.515	4.255	4.022	3.813	3.623	3.451	3.294	3.149	3.016	2.894	2.780	2.675	2.577
180	4.893	4.596	4.332	4.096	3.883	3.690	3.515	3.355	3.209	3.074	2.949	2.834	2.727	2.627
190	4.978	4.677	4.408	4.168	3.952	3.757	3.579	3.417	3.268	3.131	3.004	2.887	2.779	2.677
200	5.063	4.757	4.484	4.241	4.021	3.823	3.642	3.477	3.326	3.187	3.059	2.940	2.830	2.727
210	5.147	4.836	4.560	4.313	4.090	3.888	3.705	3.538	3.385	3.243	3.113	2.992	2.880	2.776
220	5.231	4.916	4.635	4.384	4.158	3.954	3.768	3.598	3.442	3.299	3.167	3.044	2.931	2.825
230	5.315	4.995	4.710	4.455	4.226	4.019	3.830	3.658	3.500	3.354	3.220	3.096	2.981	2.873
240	5.398	5.074	4.785	4.526	4.294	4.083	3.892	3.717	3.557	3.409	3.273	3.147	3.030	2.921
250	5.482	5.152	4.859	4.597	4.361	4.148	3.954	3.776	3.614	3.464	3.326	3.198	3.080	2.969
260	5.565	5.230	4.933	4.668	4.428	4.212	4.015	3.835	3.671	3.519	3.379	3.249	3.129	3.017
270	5.648	5.308	5.007	4.738	4.495	4.276	4.076	3.894	3.727	3.573	3.431	3.300	3.179	3.064
280	5.730	5.386	5.081	4.808	4.562	4.340	4.137	3.952	3.783	3.627	3.483	3.350	3.227	3.111
290	5.812	5.464	5.155	4.878	4.629	4.403	4.198	4.011	3.839	3.681	3.535	3.400	3.275	3.158
300	5.894	5.541	5.228	4.947	4.695	4.466	4.259	4.069	3.895	3.735	3.587	3.450	3.323	3.205
320	—	—	—	—	—	—	—	—	—	3.842	3.690	3.550	3.420	3.298

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb) (Concluded)

Temp, F	Pressure in psia									Saturation temperature in italics, F							
	145	150	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
	76.83	*78.81*	*82.64*	*86.29*	*89.78*	*93.13*	*96.34*	*99.43*	*102.42*	*105.30*	*108.09*	*110.80*	*113.42*	*115.97*	*118.45*	*120.86*	*123.21*
(*at sat*)	(*2.061*)	(*1.994*)	(*1.872*)	(*1.764*)	(*1.667*)	(*1.581*)	(*1.502*)	(*1.431*)	(*1.367*)	(*1.307*)	(*1.253*)	(*1.202*)	(*1.155*)	(*1.112*)	(*1.072*)	(*1.034*)	(*0.999*)
80	2.080	2.001	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
90	2.141	2.061	1.914	1.784	1.668	—	—	—	—	—	—	—	—	—	—	—	—
100	2.200	2.118	1.969	1.837	1.720	1.615	1.520	1.433	—	—	—	—	—	—	—	—	—
110	2.257	2.174	2.023	1.889	1.770	1.663	1.567	1.480	1.400	1.328	1.261	—	—	—	—	—	—
120	2.313	2.228	2.075	1.939	1.818	1.710	1.612	1.524	1.443	1.370	1.302	1.240	1.182	1.128	1.078	—	—
130	2.368	2.281	2.125	1.988	1.865	1.755	1.656	1.566	1.485	1.410	1.342	1.278	1.220	1.166	1.115	1.068	1.023
140	2.421	2.334	2.175	2.035	1.910	1.799	1.698	1.608	1.525	1.449	1.380	1.316	1.257	1.202	1.151	1.103	1.058
150	2.474	2.385	2.224	2.081	1.955	1.842	1.740	1.648	1.564	1.487	1.416	1.352	1.292	1.236	1.184	1.136	1.091
160	2.526	2.435	2.272	2.127	1.999	1.884	1.780	1.687	1.601	1.524	1.452	1.386	1.326	1.269	1.217	1.168	1.123
170	2.577	2.485	2.319	2.172	2.042	1.925	1.820	1.725	1.638	1.559	1.487	1.420	1.359	1.302	1.249	1.199	1.153
180	2.627	2.534	2.365	2.216	2.084	1.966	1.859	1.762	1.675	1.594	1.521	1.453	1.391	1.333	1.279	1.229	1.183
190	2.677	2.583	2.411	2.260	2.126	2.005	1.897	1.799	1.710	1.629	1.554	1.486	1.422	1.364	1.309	1.259	1.211
200	2.727	2.631	2.457	2.303	2.167	2.045	1.935	1.836	1.745	1.663	1.587	1.518	1.453	1.394	1.339	1.287	1.239
210	2.776	2.679	2.502	2.346	2.208	2.084	1.972	1.872	1.780	1.696	1.619	1.549	1.484	1.423	1.367	1.315	1.267
220	2.825	2.726	2.547	2.389	2.248	2.123	2.009	1.907	1.814	1.729	1.651	1.580	1.514	1.452	1.396	1.343	1.294
230	2.873	2.773	2.591	2.431	2.288	2.161	2.046	1.942	1.848	1.762	1.683	1.610	1.543	1.481	1.424	1.370	1.320
240	2.921	2.820	2.635	2.473	2.328	2.199	2.082	1.977	1.881	1.794	1.714	1.640	1.572	1.509	1.451	1.397	1.346
250	2.969	2.866	2.679	2.514	2.467	2.236	2.118	2.011	1.914	1.826	1.745	1.670	1.601	1.537	1.478	1.423	1.372
260	3.017	2.912	2.723	2.555	2.407	2.274	2.154	2.046	1.947	1.857	1.775	1.699	1.630	1.565	1.505	1.449	1.397
270	3.064	2.958	2.766	2.596	2.446	2.311	2.189	2.080	1.980	1.889	1.805	1.729	1.658	1.592	1.532	1.475	1.422
280	3.111	3.004	2.809	2.637	2.484	2.348	2.225	2.113	2.012	1.920	1.835	1.758	1.686	1.620	1.558	1.501	1.447
290	3.158	3.049	2.852	2.678	2.523	2.384	2.260	2.147	2.044	1.951	1.865	1.786	1.714	1.646	1.584	1.526	1.472
300	3.205	3.095	2.895	2.718	2.561	2.421	2.295	2.180	2.076	1.982	1.895	1.815	1.741	1.673	1.610	1.551	1.496
320	3.298	3.185	2.980	2.798	2.637	2.493	2.364	2.246	2.140	2.043	1.954	1.872	1.796	1.726	1.661	1.601	1.544
340	—	—	3.064	2.878	2.713	2.565	2.432	2.312	2.203	2.103	2.012	1.928	1.850	1.778	1.712	1.650	1.592
360	—	—	—	—	—	—	2.500	2.377	2.265	2.163	2.069	1.983	1.904	1.830	1.762	1.698	1.639
380	—	—	—	—	—	—	2.568	2.442	2.327	2.222	2.126	2.038	1.957	1.881	1.811	1.747	1.686
400	—	—	—	—	—	—	—	—	—	—	—	2.093	2.009	1.932	1.861	1.794	1.732

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb)

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	−63.11	*−57.64*	*−52.88*	*−48.64*	*−44.38*	*−41.34*	*−38.14*	*−35.16*	*−32.37*	*−29.76*	*−27.29*	*−24.95*	*−22.73*	*−20.61*
(at sat)	*(588.3)*	*(590.6)*	*(592.5)*	*(594.2)*	*(595.7)*	*(597.1)*	*(598.3)*	*(599.4)*	*(600.5)*	*(601.4)*	*(602.4)*	*(603.2)*	*(604.0)*	*(604.8)*
−50	595.2	594.6	594.0	—	—	—	—	—	—	—	—	—	—	—
−40	600.3	599.8	599.3	598.8	598.3	597.8	—	—	—	—	—	—	—	—
−30	605.4	604.9	604.5	604.1	603.6	603.2	602.7	602.3	601.8	—	—	—	—	—
−20	610.4	610.0	609.6	609.3	608.9	608.5	608.1	607.7	607.2	606.8	606.4	606.0	605.6	605.1
−10	615.4	615.1	614.7	614.4	614.0	613.7	613.3	613.0	612.6	612.2	611.9	611.5	611.1	610.7
0	620.4	620.1	619.8	619.5	619.2	618.9	618.5	618.2	617.9	617.6	617.2	616.9	616.6	616.2
10	625.4	625.2	624.9	624.6	624.3	624.0	623.7	623.4	623.1	622.8	622.5	622.2	621.9	621.6
20	630.4	630.2	629.9	629.7	629.4	629.1	628.9	628.6	628.3	628.0	627.8	627.5	627.2	626.9
30	635.4	635.2	635.0	634.7	634.5	634.2	634.0	633.7	633.5	633.2	633.0	632.7	632.5	632.2
40	640.4	640.2	640.0	639.8	639.5	639.3	639.1	638.9	638.6	638.4	638.2	638.0	637.7	637.5
50	645.5	645.2	645.0	644.8	644.6	644.4	644.2	644.0	643.8	643.6	643.4	643.2	642.9	642.7
60	650.5	650.3	650.1	649.9	649.7	649.5	649.3	649.1	648.9	648.7	648.5	648.3	648.1	647.9
70	655.5	655.3	655.2	655.0	654.8	654.6	654.4	654.3	654.1	653.9	653.7	653.5	653.3	653.1
80	660.6	660.4	660.2	660.1	659.9	659.7	659.6	659.4	659.2	659.0	658.9	658.7	658.5	658.4
90	665.6	665.5	665.3	665.2	665.0	664.8	664.7	664.5	664.4	664.2	664.0	663.9	663.7	663.6
100	670.7	670.6	670.4	670.3	670.1	670.0	669.8	669.7	669.5	669.4	669.2	669.1	668.9	668.8
110	675.8	675.7	675.5	675.4	675.3	675.1	675.0	674.8	674.7	674.5	674.4	674.3	674.1	674.0
120	680.9	680.8	680.7	680.5	680.4	680.3	680.1	680.0	679.9	679.7	679.6	679.5	679.3	679.2
130	686.1	685.9	685.8	685.7	685.6	685.4	685.3	685.2	685.1	684.9	684.8	684.7	684.5	684.4
140	691.2	691.1	691.0	690.9	690.7	690.6	690.5	690.4	690.3	690.1	690.0	689.9	689.8	689.7
150	696.4	696.3	696.2	696.1	695.9	695.8	695.7	695.6	595.5	695.4	695.3	695.1	695.0	694.9
160	701.6	701.5	701.4	701.3	701.2	701.1	700.9	700.8	700.7	700.6	700.5	700.4	700.3	700.2
170	706.8	706.7	706.6	706.5	706.4	706.3	706.2	706.1	706.0	705.9	705.8	705.7	705.6	705.5
180	712.1	712.0	711.9	711.8	711.7	711.6	711.5	711.4	711.3	711.2	711.1	711.0	710.9	710.8
190	—	—	—	—	—	716.9	716.8	716.7	716.6	716.5	716.4	716.3	716.2	716.1
200	—	—	—	—	—	722.2	722.1	722.0	721.9	721.8	721.7	721.6	721.5	721.4
220	—	—	—	—	—	—	—	—	—	—	—	732.3	732.2	732.2

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	−20.61	*−18.58*	*−16.64*	*−14.78*	*−12.98*	*−11.25*	*−9.58*	*−7.96*	*−6.39*	*−4.87*	*−3.40*	*−1.97*	*−0.57*	*+0.79*
(at sat)	*(604.8)*	*(605.5)*	*(606.2)*	*(606.8)*	*(607.4)*	*(608.1)*	*(608.6)*	*(609.1)*	*(609.7)*	*(610.2)*	*(610.7)*	*(611.1)*	*(611.6)*	*(612.0)*
−20	605.1	—	—	—	—	—	—	—	—	—	—	—	—	—
−10	610.7	610.3	610.0	609.6	609.2	608.8	—	—	—	—	—	—	—	—
0	616.2	615.9	615.5	615.2	614.8	614.5	614.1	613.8	613.4	613.0	612.7	612.3	611.9	—
10	621.6	621.3	621.0	620.7	620.4	620.0	619.7	619.4	619.1	618.8	618.4	618.1	617.8	617.4
20	626.9	626.7	626.4	626.1	625.8	625.5	625.2	625.0	624.7	624.4	624.1	623.8	623.5	623.2
30	632.2	632.0	631.7	631.5	631.2	630.9	630.7	630.4	630.2	629.9	629.6	629.4	629.1	628.8
40	637.5	637.3	637.0	636.8	636.6	636.3	636.1	635.8	635.6	635.4	635.1	634.9	634.6	634.4
50	642.7	642.5	642.3	642.1	641.9	641.6	641.4	641.2	641.0	640.8	640.5	640.3	640.1	639.9
60	647.9	647.7	647.5	647.3	647.1	646.9	646.7	646.5	646.3	646.1	645.9	645.7	645.5	645.3
70	653.1	653.0	652.8	652.6	652.4	652.2	652.0	651.8	651.6	651.5	651.2	651.1	650.9	650.7
80	658.4	658.2	658.0	657.8	657.7	657.5	657.3	657.1	656.9	656.8	656.6	656.4	656.2	656.1
90	663.6	663.4	663.2	663.1	662.9	662.7	662.6	662.4	662.2	662.1	661.9	661.7	661.6	661.4
100	668.8	668.6	668.5	668.3	668.1	668.0	667.8	667.7	667.5	667.4	667.2	667.1	666.9	666.7
110	674.0	673.8	673.7	673.5	673.4	673.2	673.1	673.0	672.8	672.7	672.5	672.4	672.2	672.1
120	679.2	679.1	678.9	678.8	678.6	678.5	678.4	678.2	678.1	678.0	677.8	677.7	677.5	677.4
130	684.4	684.3	684.9	684.0	683.9	683.8	683.6	683.5	683.4	683.3	683.1	683.0	682.9	682.7
140	689.7	689.5	689.4	689.3	689.2	689.0	688.9	688.8	688.7	688.6	688.4	688.3	688.2	688.1
150	694.9	694.8	694.7	694.6	694.4	694.3	694.2	694.1	694.0	693.9	693.7	693.6	693.5	693.4
160	700.2	700.1	700.0	699.8	699.7	699.6	699.5	699.4	699.3	699.2	699.1	699.0	698.8	698.7
170	705.5	705.4	705.3	705.1	705.0	704.9	704.8	704.7	704.6	704.5	704.4	704.3	704.2	704.1
180	710.8	710.7	710.6	710.5	710.4	710.3	710.2	710.1	710.0	709.9	709.8	709.7	709.6	709.5
190	716.1	716.0	715.9	715.8	715.7	715.6	715.5	715.4	715.3	715.2	715.1	715.0	714.9	714.8
200	721.4	721.3	721.2	721.1	721.1	721.0	720.9	720.8	720.7	720.6	720.5	720.4	720.3	720.2
220	732.2	732.1	732.0	731.9	731.8	731.7	731.7	731.6	731.5	731.4	731.3	731.2	731.1	731.1
240	—	—	742.8	742.7	742.7	742.6	742.6	742.5	742.4	742.3	742.2	742.2	742.0	742.0
260	—	—	—	—	—	—	—	753.4	753.3	753.2	753.2	753.1	753.0	752.9
280	—	—	—	—	—	—	—	—	—	—	—	—	764.1	764.0

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb) (Continued)

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	31	32	33	34	35	36	37	38	39	40	42	44	46	48
	0.79	*2.11*	*3.40*	*4.66*	*5.89*	*7.09*	*8.27*	*9.42*	*10.55*	*11.66*	*13.81*	*15.88*	*17.87*	*19.80*
(at sat)	*(612.0)*	*(612.4)*	*(612.8)*	*(613.2)*	*(613.6)*	*(614.0)*	*(614.3)*	*(614.7)*	*(615.0)*	*(615.4)*	*(616.0)*	*(616.6)*	*(617.2)*	*(617.7)*
10	617.4	617.1	616.8	616.4	616.1	615.7	615.4	615.0	—	—	—	—	—	—
20	623.2	622.9	622.6	622.3	622.0	621.7	621.4	621.0	620.7	620.4	619.8	619.1	618.5	617.8
30	628.8	628.5	628.3	628.0	627.7	627.4	627.2	626.9	626.6	626.3	625.8	625.2	624.6	624.0
40	634.4	634.1	633.9	633.6	633.4	633.1	632.9	632.6	632.4	632.1	631.6	631.1	630.5	630.0
50	639.9	639.6	639.4	639.2	638.9	638.7	638.5	638.3	638.0	637.8	637.3	636.8	636.4	635.9
60	645.3	645.1	644.9	644.7	644.4	644.2	644.0	643.8	643.6	643.4	643.0	642.5	642.1	741.6
70	650.7	650.5	650.3	650.1	649.9	649.7	649.5	649.3	649.1	648.9	648.5	648.1	647.7	647.3
80	565.1	655.9	655.7	655.5	655.3	655.2	655.0	654.8	654.6	654.4	654.1	653.7	653.3	652.9
90	661.4	661.2	661.1	660.9	660.7	660.6	660.4	660.2	660.1	659.9	659.5	659.2	658.9	658.5
100	666.7	666.6	666.4	666.3	666.1	666.0	665.8	665.6	665.5	665.3	665.0	664.7	664.4	664.0
110	672.1	671.9	671.8	671.6	671.5	671.3	671.2	671.0	670.9	670.7	670.4	670.1	669.8	669.5
120	677.4	677.3	667.1	677.0	676.8	676.7	676.6	676.4	676.3	676.1	675.9	675.6	675.3	675.0
130	682.7	682.6	682.5	682.3	682.2	682.1	681.9	681.8	681.7	681.5	681.3	681.0	680.7	680.5
140	688.1	687.9	687.8	687.7	687.6	687.4	687.3	687.2	687.1	686.9	686.7	686.4	686.2	685.9
150	693.4	693.3	693.2	693.0	692.9	692.8	692.7	692.6	692.5	692.3	692.1	691.9	691.6	691.4
160	698.7	698.6	698.5	698.4	698.3	698.2	698.1	698.0	697.8	697.7	697.5	697.3	697.1	696.8
170	704.1	704.0	703.9	703.8	703.7	703.6	703.5	703.3	703.2	703.1	702.9	702.7	702.5	702.3
180	709.5	709.4	709.3	709.2	709.1	709.0	708.9	708.7	708.6	708.5	708.3	708.1	707.9	707.7
190	714.8	714.7	714.6	714.5	714.5	714.4	714.3	714.2	714.1	714.0	713.8	713.6	713.4	713.2
200	720.2	720.1	720.0	720.0	719.9	719.8	719.7	719.6	719.5	719.4	719.2	719.0	718.8	718.7
220	731.1	731.0	730.9	730.8	730.7	730.6	730.6	730.5	730.4	730.3	730.1	730.0	729.8	729.6
240	742.0	741.9	741.8	741.7	741.7	741.6	741.5	741.4	741.3	741.3	741.1	741.0	740.8	740.6
260	752.9	752.9	752.8	752.7	752.7	752.6	752.5	752.4	752.4	752.3	752.2	752.0	751.9	751.7
280	764.0	763.9	763.9	763.8	763.7	763.7	763.6	763.5	763.5	763.4	763.3	763.1	763.0	762.9
300	—	—	—	—	—	—	—	—	—	774.6	774.5	774.3	774.2	774.1

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	48	50	52	54	56	58	60	62	64	66	68	70	75	80
	19.80	*21.67*	*23.48*	*25.23*	*26.94*	*28.59*	*30.21*	*31.78*	*33.31*	*34.81*	*36.27*	*37.70*	*41.13*	*44.40*
(at sat)	*(617.7)*	*(618.2)*	*(618.7)*	*(619.2)*	*(619.7)*	*(620.1)*	*(620.5)*	*(620.9)*	*(621.3)*	*(621.7)*	*(622.0)*	*(622.4)*	*(623.2)*	*(624.0)*
20	617.8	—	—	—	—	—	—	—	—	—	—	—	—	—
30	624.0	623.4	622.8	622.2	621.6	621.0	—	—	—	—	—	—	—	—
40	630.0	629.5	629.0	628.4	627.9	627.3	626.8	626.2	625.6	625.1	624.5	623.9	—	—
50	635.9	635.4	634.9	634.4	633.9	633.4	632.9	632.4	631.9	631.4	630.9	630.4	629.1	627.7
60	641.6	641.2	640.8	640.3	639.9	639.4	639.0	638.5	638.0	637.6	637.1	636.6	635.5	634.3
70	647.3	646.9	646.5	646.1	645.7	645.3	644.9	644.4	644.0	643.6	643.2	642.7	641.7	640.6
80	652.9	652.6	652.2	651.8	651.4	651.1	650.7	650.3	649.9	649.5	649.1	648.7	647.7	646.7
90	658.5	658.2	657.8	657.5	657.1	656.8	656.4	656.0	655.7	655.3	655.0	645.6	653.7	652.8
100	664.0	663.7	663.4	663.1	662.7	662.4	662.1	661.7	661.4	661.1	660.7	660.6	659.6	658.7
110	669.5	669.2	668.9	668.6	668.3	668.0	667.7	667.4	667.1	666.8	666.5	666.1	665.4	664.6
120	675.0	674.7	674.4	674.2	673.9	673.6	673.3	673.0	672.7	672.4	672.1	671.8	671.1	670.4
130	680.5	680.2	679.9	679.7	679.4	679.1	678.9	678.6	678.3	678.0	677.8	677.5	676.8	676.1
140	685.9	685.7	685.4	685.2	684.9	684.7	684.4	684.2	683.9	683.6	683.4	683.1	682.5	681.8
150	691.4	691.1	690.9	690.7	690.4	690.2	689.9	689.7	689.5	689.2	689.0	688.7	688.1	687.5
160	696.8	696.6	696.4	696.1	695.9	695.7	695.5	695.2	695.0	694.8	694.5	694.3	693.7	693.2
170	702.3	702.1	701.8	701.6	701.4	701.2	701.0	700.8	700.5	700.3	700.1	699.9	699.3	698.8
180	707.3	707.5	707.3	707.1	706.9	706.7	706.5	706.3	706.1	705.9	705.7	705.5	704.9	704.4
190	713.2	713.0	712.8	712.6	712.4	712.2	712.0	711.8	711.6	711.4	711.2	711.0	710.5	710.0
200	718.7	718.5	718.3	718.1	717.9	717.7	717.5	717.3	717.2	717.0	716.8	716.6	716.1	715.6
210	724.2	724.0	723.8	723.6	723.4	723.2	723.1	722.9	722.7	722.5	722.3	722.2	721.7	721.3
220	729.6	729.4	729.3	729.1	728.9	728.8	728.6	728.4	728.3	728.1	727.9	727.7	727.3	726.9
230	735.1	735.0	734.8	734.7	734.5	734.4	734.1	734.0	733.8	733.7	733.5	733.3	732.9	732.5
240	740.6	740.5	740.3	740.2	740.0	739.9	739.7	739.5	739.4	739.2	739.1	738.9	738.5	738.1
250	746.2	745.6	745.9	745.8	745.6	745.5	745.3	745.1	745.0	744.8	744.7	744.5	744.1	743.8
260	751.7	751.6	751.4	751.3	751.1	751.0	750.9	750.7	750.6	750.4	750.3	750.1	749.8	749.4
270	757.3	757.2	756.0	756.9	756.7	756.6	756.5	756.3	756.2	756.1	756.8	757.8	755.5	755.1
280	762.9	762.7	762.6	762.5	762.3	762.2	762.1	761.9	761.8	761.7	761.5	761.4	761.1	760.7
290	768.5	768.4	768.2	768.1	768.0	767.8	767.7	767.6	767.5	767.4	767.2	767.1	766.8	766.4
300	774.1	774.0	773.8	773.7	773.6	773.5	773.3	773.2	773.1	773.0	772.8	772.7	772.4	772.1

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb) (Continued)

Temp, F	Pressure in psia							Saturation temperature in italics, F						
	80	85	90	95	100	105	110	115	120	125	130	135	140	145
	44.40	*47.60*	*50.47*	*53.32*	*56.05*	*58.67*	*61.21*	*63.65*	*66.02*	*68.31*	*70.53*	*72.69*	*74.79*	*76.83*
(*at sat*)	(*624.0*)	(*624.7*)	(*625.3*)	(*625.9*)	(*626.5*)	(*627.0*)	(*627.5*)	(*628.0*)	(*628.4*)	(*628.8*)	(*629.2*)	(*629.6*)	(*629.9*)	(*630.2*)
50	627.7	626.4	—	—	—	—	—	—	—	—	—	—	—	—
60	634.3	633.0	631.8	630.5	629.3	627.9	—	—	—	—	—	—	—	—
70	640.6	639.5	638.3	637.2	636.0	634.9	633.7	632.5	631.3	630.0	—	—	—	—
80	646.7	645.7	644.7	643.6	642.6	641.5	640.5	639.4	638.3	637.2	636.0	634.9	633.8	632.6
90	652.8	651.8	650.9	649.9	649.0	648.0	647.0	646.0	645.0	644.0	643.0	642.0	640.9	639.9
100	658.7	657.8	657.0	656.1	655.2	654.3	653.4	652.5	651.6	650.7	649.7	648.8	647.8	646.9
110	664.6	663.8	663.0	662.1	661.3	660.5	659.7	658.8	658.0	657.1	656.3	654.4	654.5	653.6
120	670.4	669.6	668.9	668.1	667.3	666.6	665.8	665.0	664.2	663.5	662.7	661.9	661.1	660.2
130	676.1	675.4	674.7	674.0	673.3	672.6	671.9	671.1	670.4	669.7	668.9	668.2	667.4	666.7
140	681.8	681.2	680.5	679.8	679.2	678.5	677.8	677.2	676.5	675.8	675.1	674.4	673.7	673.0
150	687.5	686.9	686.3	685.6	685.0	684.4	683.7	683.1	682.5	681.8	681.2	680.5	679.9	679.2
160	693.2	692.6	692.0	691.4	690.8	690.2	689.6	689.0	688.4	687.8	687.2	686.6	686.0	685.4
170	698.8	698.2	697.7	697.1	696.6	696.0	695.4	694.9	694.3	693.7	693.2	692.6	692.0	691.4
180	704.4	703.9	703.4	702.8	702.3	701.8	701.2	700.7	700.2	699.6	699.1	698.6	698.0	697.5
190	710.0	709.5	709.0	708.5	708.0	707.5	707.0	706.5	706.0	705.5	705.0	704.5	704.0	703.4
200	715.6	715.2	714.7	714.2	713.7	713.3	712.8	712.3	711.8	711.3	710.9	710.4	709.9	709.4
210	721.3	720.8	720.4	719.9	719.4	719.0	718.5	718.1	717.6	717.2	716.7	716.2	715.8	715.3
220	726.9	726.4	726.0	725.6	725.1	724.7	724.3	723.8	723.4	723.0	722.5	722.1	721.6	721.2
230	732.5	732.1	731.7	731.3	730.8	730.4	730.0	729.6	729.2	728.8	728.3	727.9	727.5	727.1
240	738.1	737.7	737.3	736.9	736.5	736.1	735.7	735.3	734.9	734.5	734.1	733.7	733.3	732.9
250	743.8	743.4	743.0	742.6	742.2	741.9	741.5	741.1	740.7	740.3	739.9	739.6	739.2	738.8
260	749.4	749.0	748.7	748.3	747.9	747.6	747.2	746.8	746.5	746.1	745.7	745.4	745.0	744.6
270	755.1	754.7	754.4	754.0	753.6	753.3	752.9	752.6	752.2	751.9	751.5	751.2	750.8	750.5
280	760.7	760.4	760.0	759.7	759.4	759.0	758.7	758.4	758.0	757.7	757.3	757.0	756.7	756.3
290	766.4	766.1	765.8	765.8	765.1	764.8	764.5	764.1	763.8	763.5	763.1	762.8	762.5	762.2
300	772.1	771.8	771.5	771.2	770.8	770.5	770.2	769.9	769.6	769.3	769.0	768.6	768.3	768.0
320	—	—	—	—	—	—	—	—	—	780.9	780.6	780.3	780.0	779.7

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb) (Concluded)

Temp, F	Pressure in psia											Saturation temperature in italics, F					
	145	150	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
	76.83	*78.81*	*82.64*	*86.29*	*89.78*	*93.13*	*96.34*	*99.43*	*102.42*	*105.30*	*108.09*	*110.80*	*113.42*	*115.97*	*118.45*	*120.86*	*123.21*
at sat)	(*630.2*)	(*630.5*)	(*631.1*)	(*631.6*)	(*632.0*)	(*632.4*)	(*632.7*)	(*633.0*)	(*633.2*)	(*633.4*)	(*633.6*)	(*633.8*)	(*633.9*)	(*633.9*)	(*634.0*)	(*634.0*)	(*634.0*)
80	632.6	631.4	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
90	639.9	638.8	636.6	634.4	632.2	—	—	—	—	—	—	—	—	—	—	—	—
100	646.9	645.9	643.9	641.9	639.9	637.8	635.6	633.5	—	—	—	—	—	—	—	—	—
110	653.6	652.8	651.0	649.1	647.3	645.4	643.4	641.5	639.4	637.4	635.3	—	—	—	—	—	—
120	660.2	659.4	657.8	656.1	654.4	652.6	650.9	649.1	647.3	645.4	643.5	641.5	639.5	637.5	635.4	—	—
130	666.7	665.9	664.4	662.8	661.3	659.7	658.1	656.4	654.8	653.1	651.3	649.6	647.8	645.9	644.0	642.1	640.1
140	673.0	672.3	670.9	669.4	668.0	666.5	665.0	663.5	662.0	660.4	658.8	657.2	655.6	653.9	652.2	650.5	648.7
150	679.2	678.6	677.2	675.9	674.6	673.2	671.8	670.4	669.0	667.6	666.1	664.6	663.1	661.6	660.1	658.5	656.9
160	685.4	684.8	683.5	682.3	681.0	679.7	678.4	677.1	675.8	674.5	673.1	671.8	670.4	669.0	667.6	666.1	664.7
170	691.4	690.9	689.7	688.5	687.3	686.1	684.9	683.7	682.5	681.3	680.0	678.7	677.5	676.2	674.9	673.5	672.2
180	697.5	696.9	695.8	694.7	693.6	692.5	691.3	690.2	689.1	687.9	686.7	685.5	684.4	683.2	681.9	680.7	679.5
190	703.4	702.9	701.9	700.8	699.8	698.7	697.7	696.6	695.5	694.4	693.3	692.2	691.1	690.0	688.9	687.7	686.5
200	709.4	708.9	707.9	706.9	705.9	704.9	703.9	702.9	701.9	700.9	699.8	698.8	697.7	696.7	695.6	694.6	693.5
210	715.3	714.8	713.9	713.0	712.0	711.1	710.1	709.2	708.2	707.2	706.2	705.3	704.3	703.3	702.3	701.3	700.3
220	721.2	720.7	719.9	719.0	718.1	717.2	716.3	715.3	714.4	713.5	712.6	711.7	710.7	709.8	708.8	707.9	706.9
230	727.1	726.6	725.8	724.9	724.1	723.2	722.4	721.5	720.6	719.8	718.9	718.0	717.1	716.2	715.3	714.4	713.5
240	732.9	732.5	731.7	730.9	730.1	729.3	728.4	727.6	726.8	726.0	725.1	724.3	723.4	722.6	721.8	720.9	720.0
250	738.8	738.4	737.6	736.8	736.1	735.3	734.5	733.7	732.9	732.1	731.3	730.5	729.7	728.9	728.1	727.3	726.5
260	744.6	744.3	743.5	742.8	742.0	741.3	740.5	739.8	739.0	738.3	737.5	736.7	736.0	735.2	734.4	733.7	732.9
270	750.5	750.1	749.4	748.7	748.0	747.3	746.5	745.8	745.1	744.4	743.0	742.9	742.2	741.4	740.7	740.0	739.2
280	756.3	756.0	755.3	754.6	753.9	753.2	752.5	751.8	751.1	750.5	749.8	749.1	748.4	747.7	747.0	746.3	745.5
290	762.2	761.8	761.2	760.5	759.9	759.2	758.5	757.9	757.2	756.5	755.9	755.2	754.5	753.9	753.2	752.5	751.8
300	768.0	767.7	767.1	766.4	765.8	765.2	764.5	763.9	763.2	762.6	762.0	761.3	760.7	760.0	759.4	758.7	758.1
320	779.7	779.4	778.9	778.3	777.7	777.1	776.5	775.9	775.3	774.7	774.1	773.5	772.9	772.3	771.7	771.1	770.5
340	—	791.2	790.7	790.1	789.6	789.0	788.5	787.9	787.4	786.8	786.3	785.7	785.2	784.6	784.0	783.5	782.9
360	—	—	—	—	—	—	800.5	800.0	799.5	798.9	798.4	797.9	797.4	796.9	796.3	795.8	795.3
380	—	—	—	—	—	—	812.5	812.0	811.6	811.1	810.6	810.1	809.6	809.1	808.7	808.2	807.7
400	—	—	—	—	—	—	—	—	—	—	—	822.3	821.9	821.4	821.0	820.5	820.1

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F)

Temp, F	Pressure in psia							Saturation temperature in italics, F						
	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	−63.11	*−57.64*	*−52.88*	*−48.64*	*−44.38*	*−41.34*	*−38.14*	*−35.16*	*−32.37*	*−29.76*	*−27.29*	*−24.96*	*−22.73*	*−20.61*
(*at sat*)	(*1.4857*)	(*1.4703*)	(*1.4574*)	(*1.4462*)	(*1.4363*)	(*1.4276*)	(*1.4196*)	(*1.4124*)	(*1.4057*)	(*1.3996*)	(*1.3938*)	(*1.3885*)	(*1.3835*)	(*1.3787*)
−50	1.5025	1.4803	1.4611	—	—	—	—	—	—	—	—	—	—	—
−40	1.5149	1.4928	1.4739	1.4573	1.4426	1.4293	—	—	—	—	—	—	—	—
−30	1.5269	1.5049	1.4861	1.4697	1.4551	1.4420	1.4300	1.4190	1.4088	—	—	—	—	—
−20	1.5385	1.5166	1.4979	1.4816	1.4672	1.4542	1.4423	1.4314	1.4213	1.4119	1.4031	1.3948	1.3870	1.3795
−10	1.5498	1.5280	1.5094	1.4932	1.4788	1.4659	1.4542	1.4434	1.4334	1.4241	1.4154	1.4072	1.3994	1.3921
0	1.5608	1.5391	1.5206	1.5044	1.4902	1.4773	1.4656	1.4549	1.4450	1.4358	1.4272	1.4191	1.4114	1.4042
10	1.5716	1.5499	1.5314	1.5154	1.5012	1.4884	1.4768	1.4661	1.4563	1.4472	1.4386	1.4306	1.4230	1.4158
20	1.5821	1.5605	1.5421	1.5261	1.5119	1.4992	1.4876	1.4770	1.4672	1.4582	1.4497	1.4417	1.4342	1.4270
30	1.5925	1.5708	1.5525	1.5365	1.5224	1.5097	1.4982	1.4877	1.4779	1.4688	1.4604	1.4525	1.4450	1.4380
40	1.6026	1.5810	1.5627	1.5467	1.5327	1.5200	1.5085	1.4980	1.4883	1.4793	1.4709	1.4630	1.4556	1.4486
50	1.6125	1.5910	1.5727	1.5568	1.5427	1.5301	1.5187	1.5082	1.4985	1.4896	1.4812	1.4733	1.4659	1.4590
60	1.6223	1.6008	1.5825	1.5666	1.5526	1.5400	1.5286	1.5182	1.5085	1.4996	1.4912	1.4834	1.4761	1.4691
70	1.6319	1.6104	1.5921	1.5763	1.5623	1.5497	1.5383	1.5279	1.5183	1.5094	1.5011	1.4933	1.4860	1.4790
80	1.6413	1.6199	1.6016	1.5858	1.5718	1.5593	1.5479	1.5375	1.5279	1.5191	1.5108	1.5030	1.4957	1.4887
90	1.6506	1.6292	1.6110	1.5952	1.5812	1.5687	1.5573	1.5470	1.5374	1.5285	1.5203	1.5125	1.5052	1.4983
100	1.6598	1.6384	1.6202	1.6044	1.5904	1.5779	1.5666	1.5562	1.5467	1.5378	1.5296	1.5218	1.5146	1.5077
110	1.6689	1.6474	1.6292	1.6135	1.5995	1.5870	1.5757	1.5654	1.5558	1.5470	1.5388	1.5310	1.5238	1.5169
120	1.6778	1.6563	1.6382	1.6224	1.6085	1.5960	1.5847	1.5744	1.5649	1.5560	1.5478	1.5401	1.5328	1.5260
130	1.6865	1.6651	1.6470	1.6312	1.6173	1.6049	1.5936	1.5833	1.5737	1.5649	1.5567	1.5490	1.5418	1.5349
140	1.6952	1.6738	1.6557	1.6399	1.6260	1.6136	1.6023	1.5920	1.5825	1.5737	1.5655	1.5578	1.5506	1.5438
150	1.7038	1.6824	1.6643	1.6485	1.6346	1.6222	1.6109	1.6006	1.5911	1.5824	1.5742	1.5665	1.5593	1.5525
160	1.7122	1.6909	1.6727	1.6570	1.6431	1.6307	1.6194	1.6092	1.5997	1.5909	1.5827	1.5750	1.5678	1.5610
170	1.7206	1.6992	1.6811	1.6654	1.6515	1.6391	1.6278	1.6176	1.6081	1.5993	1.5911	1.5835	1.5763	1.5695
180	1.7289	1.7075	1.6894	1.6737	1.6598	1.6474	1.6362	1.6259	1.6164	1.6076	1.5995	1.5918	1.5846	1.5778
190	—	—	—	—	—	1.6556	1.6444	1.6341	1.6246	1.6159	1.6077	1.6001	1.5929	1.5861
200	—	—	—	—	—	1.6637	1.6525	1.6422	1.6328	1.6240	1.6158	1.6082	1.6010	1.5943
210	—	—	—	—	—	—	—	—	—	—	1.6238	1.6162	1.6090	1.5973
220	—	—	—	—	—	—	—	—	—	—	1.6318	1.6242	1.6170	1.6103

Temp, F	Pressure in psia							Saturation temperature in italics, F						
	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	−20.61	*−18.58*	*−16.64*	*−14.78*	*−12.98*	*−11.25*	*−9.58*	*−7.96*	*−6.39*	*−4.87*	*−3.40*	*−1.97*	*−0.57*	*+0.79*
(*at sat*)	(*1.3787*)	(*1.3742*)	(*1.3700*)	(*1.3659*)	(*1.3621*)	(*1.3584*)	(*1.3549*)	(*1.3515*)	(*1.3482*)	(*1.3451*)	(*1.3421*)	(*1.3392*)	(*1.3364*)	(*1.3336*)
−20	1.3795	—	—	—	—	—	—	—	—	—	—	—	—	—
−10	1.3921	1.3851	1.3784	1.3720	1.3659	1.3600	—	—	—	—	—	—	—	—
0	1.4042	1.3973	1.3907	1.3844	1.3784	1.3726	1.3670	1.3616	1.3564	1.3513	1.3465	1.3417	1.3371	—
10	1.4158	1.4090	1.4025	1.3962	1.3903	1.3846	1.3791	1.3738	1.3686	1.3637	1.3589	1.3542	1.3497	1.3453
20	1.4270	1.4203	1.4138	1.4077	1.4018	1.3961	1.3907	1.3855	1.3804	1.3755	1.3708	1.3662	1.3618	1.3574
30	1.4380	1.4312	1.4248	1.4187	1.4129	1.4073	1.4019	1.3967	1.3917	1.3869	1.3822	1.3777	1.3733	1.3691
40	1.4486	1.4419	1.4356	1.4295	1.4237	1.4181	1.4128	1.4077	1.4027	1.3979	1.3933	1.3888	1.3845	1.3803
50	1.4590	1.4523	1.4460	1.4400	1.4342	1.4287	1.4234	1.4183	1.4134	1.4087	1.4041	1.3996	1.3953	1.3912
60	1.4691	1.4625	1.4562	1.4502	1.4445	1.4390	1.4337	1.4287	1.4238	1.4191	1.4145	1.4101	1.4059	1.4017
70	1.4790	1.4724	1.4662	1.4602	1.4545	1.4491	1.4438	1.4388	1.4339	1.4292	1.4247	1.4204	1.4161	1.4120
80	1.4887	1.4822	1.4760	1.4700	1.4643	1.4589	1.4537	1.4487	1.4439	1.4392	1.4347	1.4304	1.4261	1.4221
90	1.4983	1.4918	1.4856	1.4796	1.4740	1.4686	1.4634	1.4584	1.4536	1.4489	1.4445	1.4401	1.4359	1.4319
100	1.5077	1.5012	1.4950	1.4891	1.4834	1.4780	1.4729	1.4679	1.4631	1.4585	1.4540	1.4497	1.4456	1.4415
110	1.5169	1.5104	1.5042	1.4983	1.4927	1.4873	1.4822	1.4772	1.4725	1.4679	1.4634	1.4591	1.4550	1.4510
120	1.5260	1.5195	1.5133	1.5075	1.5019	1.4965	1.4914	1.4864	1.4817	1.4771	1.4726	1.4684	1.4642	1.4602
130	1.5349	1.5285	1.5223	1.5165	1.5109	1.5055	1.5004	1.4954	1.4907	1.4861	1.4817	1.4775	1.4733	1.4693
140	1.5438	1.5373	1.5312	1.5253	1.5197	1.5144	1.5093	1.5043	1.4996	1.4950	1.4906	1.4864	1.4823	1.4783
150	1.5525	1.5460	1.5399	1.5340	1.5285	1.5231	1.5180	1.5131	1.5084	1.5038	1.4994	1.4952	1.4911	1.4871
160	1.5610	1.5546	1.5485	1.5426	1.5371	1.5317	1.5266	1.5217	1.5170	1.5125	1.5081	1.5039	1.4998	1.4958
170	1.5695	1.5631	1.5569	1.5510	1.5456	1.5402	1.5352	1.5303	1.5256	1.5210	1.5167	1.5124	1.5083	1.5044
180	1.5778	1.5714	1.5653	1.5595	1.5539	1.5486	1.5436	1.5387	1.5340	1.5295	1.5251	1.5209	1.5168	1.5129
190	1.5861	1.5797	1.5736	1.5678	1.5622	1.5569	1.5518	1.5470	1.5423	1.5378	1.5334	1.5292	1.5251	1.5212
200	1.5943	1.5878	1.5817	1.5759	1.5704	1.5651	1.5600	1.5552	1.5505	1.5460	1.5416	1.5374	1.5334	1.5294
210	1.6023	1.5959	1.5898	1.5839	1.5785	1.5732	1.5681	1.5633	1.5586	1.5541	1.5497	1.5455	1.5415	1.5375
220	1.6103	1.6039	1.5978	1.5920	1.5865	1.5812	1.5761	1.5713	1.5666	1.5621	1.5578	1.5536	1.5495	1.5456
230	—	—	1.6057	1.5999	1.5944	1.5891	1.5840	1.5792	1.5745	1.5680	1.5657	1.5615	1.5574	1.5535
240	—	—	1.6135	1.6077	1.6022	1.5969	1.5919	1.5870	1.5824	1.5779	1.5736	1.5694	1.5653	1.5614
250	—	—	—	—	—	—	—	1.5948	1.5901	1.5856	1.5813	1.5771	1.5731	1.5692
260	—	—	—	—	—	—	—	1.6025	1.5978	1.5933	1.5890	1.5848	1.5808	1.5769
270	—	—	—	—	—	—	—	—	—	—	—	—	1.5884	1.5845
280	—	—	—	—	—	—	—	—	—	—	—	—	1.5960	1.5921

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F) (Continued)

emp, F	Pressure in psia								Saturation temperature in italics, F					
	31	32	33	34	35	36	37	38	39	40	42	44	46	48
	0.79	*2.11*	*3.40*	*4.66*	*5.89*	*7.09*	*8.27*	*9.42*	*10.55*	*11.66*	*13.81*	*15.88*	*17.87*	*19.80*
t sat)	*(1.3336)*	*(1.3310)*	*(1.3286)*	*(1.3260)*	*(1.3236)*	*(1.3213)*	*(1.3190)*	*(1.3166)*	*(1.3146)*	*(1.3125)*	*(1.3084)*	*(1.3046)*	*(1.3009)*	*(1.2973)*
10	1.3453	1.3411	1.3369	1.3328	1.3289	1.3250	1.3212	1.3175	—	—	—	—	—	—
20	1.3574	1.3532	1.3492	1.3452	1.3413	1.3375	1.3338	1.3301	1.3266	1.3231	1.3164	1.3099	1.3036	1.2976
30	1.3691	1.3649	1.3609	1.3570	1.3532	1.3494	1.3458	1.3422	1.3387	1.3353	1.3287	1.3224	1.3162	1.3103
40	1.3803	1.3762	1.3722	1.3684	1.3646	1.3609	1.3573	1.3538	1.3504	1.3470	1.3405	1.3343	1.3283	1.3225
50	1.3912	1.3871	1.3832	1.3793	1.3756	1.3720	1.3684	1.3650	1.3616	1.3583	1.3519	1.3457	1.3398	1.3341
60	1.4017	1.3977	1.3938	1.3900	1.3863	1.3827	1.3792	1.3758	1.3724	1.3692	1.3628	1.3567	1.3509	1.3453
70	1.4120	1.4080	1.4042	1.4004	1.3967	1.3932	1.3897	1.3863	1.3830	1.3797	1.3734	1.3674	1.3617	1.3561
80	1.4221	1.4181	1.4143	1.4105	1.4069	1.4033	1.3999	1.3965	1.3932	1.3900	1.3838	1.3778	1.3721	1.3666
90	1.4319	1.4280	1.4241	1.4204	1.4168	1.4133	1.4098	1.4065	1.4032	1.4000	1.3939	1.3880	1.3823	1.3768
100	1.4415	1.4376	1.4338	1.4301	1.4265	1.4230	1.4196	1.4163	1.4130	1.4098	1.4037	1.3978	1.3922	1.3868
110	1.4510	1.4470	1.4433	1.4396	1.4360	1.4325	1.4291	1.4258	1.4226	1.4194	1.4133	1.4075	1.4019	1.3965
120	1.4602	1.4563	1.4526	1.4489	1.4453	1.4419	1.4385	1.4352	1.4320	1.4288	1.4228	1.4170	1.4114	1.4061
130	1.4693	1.4655	1.4617	1.4581	1.4545	1.4510	1.4477	1.4444	1.4412	1.4381	1.4320	1.4263	1.4207	1.4154
140	1.4783	1.4744	1.4707	1.4671	1.4635	1.4601	1.4567	1.4534	1.4503	1.4471	1.4411	1.4354	1.4299	1.4246
150	1.4871	1.4833	1.4795	1.4759	1.4724	1.4689	1.4656	1.4623	1.4592	1.4561	1.4501	1.4444	1.4389	1.4336
160	1.4958	1.4920	1.4883	1.4846	1.4811	1.4777	1.4744	1.4711	1.4679	1.4648	1.4589	1.4532	1.4477	1.4425
170	1.5044	1.5006	1.4968	1.4932	1.4897	1.4863	1.4830	1.4797	1.4766	1.4735	1.4676	1.4619	1.4564	1.4512
180	1.5129	1.5090	1.5053	1.5017	1.4982	1.4948	1.4915	1.4883	1.4851	1.4820	1.4761	1.4704	1.4650	1.4598
190	1.5212	1.5174	1.5137	1.5101	1.5066	1.5032	1.4999	1.4966	1.4935	1.4904	1.4845	1.4789	1.4735	1.4683
200	1.5294	1.5256	1.5219	1.5183	1.5148	1.5115	1.5082	1.5049	1.5018	1.4987	1.4928	1.4872	1.4818	1.4766
210	1.5375	1.5337	1.5300	1.5265	1.5230	1.5196	1.5163	1.5131	1.5100	1.5069	1.5010	1.4954	1.4900	1.4848
220	1.5456	1.5418	1.5381	1.5346	1.5311	1.5277	1.5244	1.5212	1.5181	1.5150	1.5091	1.5035	1.4981	1.4930
230	1.5535	1.5497	1.5461	1.5425	1.5390	1.5357	1.5324	1.5292	1.5261	1.5230	1.5171	1.5115	1.5061	1.5010
240	1.5614	1.5576	1.5540	1.5504	1.5469	1.5436	1.5403	1.5371	1.5340	1.5309	1.5251	1.5195	1.5141	1.5090
250	1.5692	1.5654	1.5619	1.5582	1.5547	1.5514	1.5481	1.5449	1.5418	1.5387	1.5329	1.5273	1.5219	1.5168
260	1.5769	1.5731	1.5695	1.5659	1.5624	1.5591	1.5558	1.5526	1.5495	1.5465	1.5406	1.5350	1.5297	1.5246
270	1.5845	1.5757	1.5721	1.5735	1.5700	1.5667	1.5634	1.5602	1.5571	1.5541	1.5483	1.5427	1.5374	1.5323
280	1.5921	1.5883	1.5846	1.5811	1.5776	1.5743	1.5710	1.5678	1.5647	1.5617	1.5559	1.5503	1.5450	1.5399
290	—	—	—	—	—	—	—	—	—	1.5692	1.5634	1.5578	1.5525	1.5474
300	—	—	—	—	—	—	—	—	—	1.5766	1.5708	1.5652	1.5599	1.5548

Temp, F	Pressure in psia								Saturation temperature in italics, F					
	48	50	52	54	56	58	60	62	64	66	68	70	75	80
	19.80	*21.67*	*23.48*	*25.23*	*26.94*	*28.59*	*30.21*	*31.78*	*33.31*	*34.81*	*36.27*	*37.70*	*41.13*	*44.40*
(at sat)	*(1.2973)*	*(1.2939)*	*(1.2906)*	*(1.2875)*	*(1.2844)*	*(1.2815)*	*(1.2787)*	*(1.2759)*	*(1.2733)*	*(1.2707)*	*(1.2682)*	*(1.2658)*	*(1.2599)*	*(1.2545)*
20	1.2976	—	—	—	—	—	—	—	—	—	—	—	—	—
30	1.3103	1.3046	1.2991	1.2937	1.2884	1.2834	—	—	—	—	—	—	—	—
40	1.3225	1.3169	1.3114	1.3062	1.3011	1.2961	1.2913	1.2866	1.2820	1.2775	1.2731	1.2688	—	—
50	1.3341	1.3268	1.3233	1.3181	1.3131	1.3082	1.3035	1.2989	1.2944	1.2900	1.2858	1.2816	1.2715	1.2619
60	1.3453	1.3399	1.3346	1.3295	1.3246	1.3199	1.3152	1.3107	1.3063	1.3020	1.2978	1.2937	1.2839	1.2745
70	1.3561	1.3508	1.3456	1.3406	1.3357	1.3310	1.3265	1.3220	1.3177	1.3135	1.3094	1.3054	1.2957	1.2866
80	1.3666	1.3613	1.3562	1.3513	1.3465	1.3418	1.3373	1.3330	1.3287	1.3245	1.3205	1.3166	1.3071	1.2981
90	1.3768	1.3716	1.3665	1.3616	1.3569	1.3523	1.3479	1.3435	1.3393	1.3352	1.3312	1.3274	1.3180	1.3092
100	1.3868	1.3816	1.3766	1.3717	1.3670	1.3625	1.3581	1.3538	1.3496	1.3456	1.3417	1.3378	1.3286	1.3199
110	1.3965	1.3914	1.3864	1.3816	1.3769	1.3724	1.3681	1.3638	1.3597	1.3557	1.3518	1.3480	1.3389	1.3303
120	1.4061	1.4009	1.3960	1.3912	1.3866	1.3821	1.3778	1.3736	1.3695	1.3655	1.3617	1.3579	1.3489	1.3404
130	1.4154	1.4103	1.4054	1.4006	1.3961	1.3916	1.3873	1.3831	1.3791	1.3751	1.3713	1.3676	1.3586	1.3502
140	1.4246	1.4195	1.4146	1.4099	1.4053	1.4009	1.3966	1.3925	1.3885	1.3846	1.3807	1.3770	1.3682	1.3598
150	1.4336	1.4286	1.4237	1.4190	1.4144	1.4100	1.4058	1.4017	1.3977	1.3938	1.3900	1.3863	1.3775	1.3692
160	1.4425	1.4374	1.4326	1.4279	1.4234	1.4190	1.4148	1.4107	1.4067	1.4028	1.3991	1.3954	1.3866	1.3784
170	1.4512	1.4462	1.4413	1.4367	1.4322	1.4278	1.4236	1.4195	1.4156	1.4117	1.4080	1.4043	1.3956	1.3874
180	1.4598	1.4548	1.4500	1.4453	1.4408	1.4365	1.4323	1.4282	1.4243	1.4205	1.4167	1.4131	1.4044	1.3963
190	1.4683	1.4633	1.4585	1.4538	1.4494	1.4450	1.4409	1.4368	1.4329	1.4291	1.4254	1.4217	1.4131	1.4050
200	1.4766	1.4716	1.4668	1.4622	1.4578	1.4535	1.4493	1.4453	1.4413	1.4375	1.4338	1.4302	1.4217	1.4136
210	1.4848	1.4799	1.4751	1.4705	1.4661	1.4618	1.4576	1.4536	1.4497	1.4459	1.4422	1.4386	1.4301	1.4220
220	1.4930	1.4880	1.4833	1.4787	1.4743	1.4700	1.4658	1.4618	1.4579	1.4541	1.4505	1.4469	1.4384	1.4304
230	1.5010	1.4960	1.4913	1.4867	1.4823	1.4780	1.4739	1.4699	1.4660	1.4623	1.4586	1.4550	1.4466	1.4386
240	1.5090	1.5040	1.4993	1.4947	1.4903	1.4860	1.4819	1.4779	1.4741	1.4703	1.4666	1.4631	1.4546	1.4467
250	1.5168	1.5119	1.5071	1.5026	1.4982	1.4939	1.4898	1.4858	1.4820	1.4782	1.4745	1.4710	1.4625	1.4547
260	1.5246	1.5197	1.5149	1.5104	1.5060	1.5017	1.4976	1.4937	1.4898	1.4861	1.4824	1.4789	1.4705	1.4626
270	1.5323	1.5274	1.5226	1.5181	1.5137	1.5094	1.5053	1.5014	1.4975	1.4938	1.4902	1.4866	1.4783	1.4704
280	1.5399	1.5350	1.5303	1.5257	1.5213	1.5171	1.5130	1.5091	1.5052	1.5015	1.4979	1.4943	1.4860	1.4781
290	1.5474	1.5425	1.5378	1.5332	1.5289	1.5246	1.5206	1.5116	1.5128	1.5091	1.5055	1.5019	1.4936	1.4857
300	1.5548	1.5500	1.5453	1.5407	1.5364	1.5321	1.5281	1.5241	1.5203	1.5166	1.5130	1.5095	1.5011	1.4933

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F) (Continued)

Temp, F	Pressure in psia							Saturation temperature in italics, F						
	80	85	90	95	100	105	110	115	120	125	130	135	140	145
	44.40	*47.60*	*50.47*	*53.32*	*56.05*	*58.67*	*61.21*	*63.65*	*66.02*	*68.31*	*70.53*	*72.69*	*74.79*	*76.83*
(at sat)	*(1.2546)*	*(1.2494)*	*(1.2446)*	*(1.2399)*	*(1.2356)*	*(1.2314)*	*(1.2276)*	*1.2237)*	*(1.2201)*	*(1.2166)*	*(1.2132)*	*(1.2100)*	*(1.2068)*	*(1.2038)*
50	1.2619	1.2527	—	—	—	—	—	—	—	—	—	—	—	—
60	1.2745	1.2656	1.2571	1.2489	1.2409	1.2332	—	—	—	—	—	—	—	—
70	1.2866	1.2779	1.2695	1.2616	1.2539	1.2464	1.2392	1.2323	1.2255	1.2189	—	—	—	—
80	1.2981	1.2896	1.2814	1.2736	1.2661	1.2589	1.2519	1.2451	1.2386	1.2322	1.2660	1.2199	1.2140	1.2082
90	1.3092	1.3008	1.2928	1.2852	1.2778	1.2708	1.2640	1.2574	1.2510	1.2448	1.2388	1.2329	1.2272	1.2216
100	1.3199	1.3117	1.3038	1.2963	1.2891	1.2822	1.2755	1.2690	1.2628	1.2568	1.2509	1.2452	1.2396	1.2342
110	1.3303	1.3221	1.3144	1.3070	1.2999	1.2931	1.2866	1.2802	1.2741	1.2682	1.2625	1.2569	1.2515	1.2462
120	1.3404	1.3323	1.3247	1.3174	1.3104	1.3037	1.2972	1.2910	1.2850	1.2792	1.2736	1.2681	1.2628	1.2577
130	1.3502	1.3422	1.3347	1.3275	1.3206	1.3139	1.3076	1.3015	1.2956	1.2899	1.2843	1.2790	1.2738	1.2687
140	1.3598	1.3519	1.3444	1.3373	1.3305	1.3239	1.3176	1.3116	1.3058	1.3002	1.2947	1.2894	1.2843	1.2793
150	1.3692	1.3614	1.3539	1.3469	1.3401	1.3336	1.3274	1.3215	1.3157	1.3102	1.3048	1.2996	1.2945	1.2896
160	1.3784	1.3706	1.3633	1.3562	1.3495	1.3431	1.3370	1.3311	1.3254	1.3199	1.3146	1.3094	1.3045	1.2996
170	1.3874	1.3797	1.3724	1.3654	1.3588	1.3521	1.3463	1.3405	1.3348	1.3294	1.3241	1.3191	1.3141	1.3093
180	1.3963	1.3886	1.3813	1.3744	1.3678	1.3615	1.3555	1.3497	1.3441	1.3387	1.3335	1.3284	1.3236	1.3188
190	1.4050	1.3974	1.3901	1.3833	1.3767	1.3704	1.3644	1.3587	1.3531	1.3478	1.3426	1.3376	1.3328	1.3281
220	1.4136	1.4060	1.3988	1.3919	1.3854	1.3792	1.3732	1.3675	1.3620	1.3567	1.3516	1.3466	1.3418	1.3372
210	1.4220	1.4145	1.4073	1.4005	1.3940	1.3878	1.3819	1.3762	1.3707	1.3654	1.3604	1.3554	1.3507	1.3461
220	1.4304	1.4228	1.4157	1.4089	1.4024	1.3963	1.3904	1.3847	1.3793	1.3740	1.3690	1.3641	1.3594	1.3548
230	1.4386	1.4311	1.4239	1.4172	1.4108	1.4046	1.3988	1.3931	1.3877	1.3825	1.3775	1.3726	1.3679	1.3634
240	1.4467	1.4392	1.4321	1.4254	1.4190	1.4129	1.4070	1.4014	1.3960	1.3908	1.3858	1.3810	1.3763	1.3718
250	1.4547	1.4472	1.4401	1.4334	1.4271	1.4210	1.4151	1.4096	1.4042	1.3990	1.3941	1.3893	1.3846	1.3801
260	1.4626	1.4551	1.4481	1.4414	1.4350	1.4290	1.4232	1.4176	1.4123	1.4071	1.4022	1.3974	1.3928	1.3883
270	1.4704	1.4629	1.4559	1.4492	1.4429	1.4369	1.4311	1.4256	1.4202	1.4151	1.4102	1.4054	1.4008	1.3964
280	1.4781	1.4707	1.4637	1.4570	1.4507	1.4447	1.4389	1.4334	1.4281	1.4230	1.4181	1.4133	1.4088	1.4043
290	1.4857	1.4783	1.4713	1.4647	1.4584	1.4524	1.4466	1.4411	1.4359	1.4308	1.4259	1.4212	1.4166	1.4122
300	1.4933	1.4859	1.4789	1.4723	1.4660	1.4600	1.4543	1.4488	1.4435	1.4385	1.4336	1.4289	1.4243	1.4199
320	—	—	—	—	—	—	—	—	—	1.4536	1.4487	1.4441	1.4395	1.4352

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F) (Concluded)

Temp, F	Pressure in psia										Saturation temperature in italics, F						
	145	150	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
	76 83	*78 81*	*82 64*	*86 29*	*89.78*	*93.13*	*96 34*	*99 43*	*102 42*	*105 90*	*108 09*	*110 80*	*113 42*	*115 97*	*118 45*	*120 86*	*123.21*
(at sat)	*(1 2038)*	*(1 2009)*	*(1.1952)*	*(1.1900)*	*(1 1850)*	*(1 1802)*	*(1 1756)*	*(1 1713)*	*(1 1671)*	*(1 1631)*	*(1 1592)*	*(1 1555)*	*(1 1518)*	*(1 1483)*	*(1 1449)*	*(1 1416)*	*(1 1383)*
80	1.2082	1.2025	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
90	1.2216	1.2161	1.2055	1.1952	1.1853	—	—	—	—	—	—	—	—	—	—	—	—
100	1.2342	1.2289	1.2186	1.2087	1.1992	1.1899	1.1809	1.1721	—	—	—	—	—	—	—	—	—
110	1.2462	1.2410	1.2311	1.2215	1.2123	1.2034	1.1947	1.1863	1.1781	1.1700	1.1621	—	—	—	—	—	—
120	1.2577	1.2526	1.2429	1.2336	1.2247	1.2160	1.2077	1.1996	1 1917	1.1840	1.1764	1.1690	1.1617	1.1544	1.1473	—	—
130	1.2687	1.2638	1.2542	1.2452	1.2364	1.2281	1.2200	1.2121	1.2045	1.1971	1.1898	1.1827	1.1757	1.1689	1.1621	1.1554	1.1487
140	1.2793	1.2745	1.2652	1.2563	1.2477	1.2396	1.2317	1.2240	1.2167	1.2095	1.2025	1.1956	1.1889	1.1823	1.1759	1.1695	1.1632
150	1.2896	1.2849	1.2757	1.2669	1.2586	1.2506	1.2429	1.2354	1.2281	1 2213	1.2145	1.2078	1.2014	1.1950	1.1888	1.1827	1.1767
160	1.2996	1.2949	1.2859	1.2773	1.2691	1.2612	1.2537	1.2464	1.2394	1.2325	1.2259	1.2195	1.2132	1.2071	1.2011	1.1952	1.1894
170	1.3093	1.3047	1.2958	1.2873	1.2792	1.2715	1.2641	1.2569	1.2501	1.2434	1.2369	1.2306	1.2245	1.2185	1.2127	1.2070	1.2014
180	1.3188	1.3142	1.3054	1.2971	1.2891	1.2815	1.2742	1.2672	1.2604	1.2538	1.2475	1.2414	1.2354	1.2296	1.2239	1.2183	1.2129
190	1.3281	1.3236	1.3148	1.3066	1.2987	1.2912	1.2840	1.2771	1.2704	1.2640	1.2577	1.2517	1.2458	1.2401	1.2346	1.2292	1.2239
200	1.3372	1.3327	1.3240	1.3159	1.3081	1.3007	1.2935	1.2867	1.2801	1.2738	1.2677	1.2617	1.2560	1.2504	1.2449	1.2396	1.2344
210	1.3461	1.3416	1.3331	1.3249	1.3172	1.3099	1.3029	1.2961	1.2896	1.2834	1.2773	1.2715	1.2658	1.2603	1.2550	1.2497	1.2447
220	1.3548	1.3504	1.3419	1.3338	1.3262	1.3189	1.3120	1.3053	1.2989	1.2927	1.2867	1.2810	1.2754	1.2700	1.2647	1.2596	1.2546
230	1.3634	1.3590	1.3506	1.3426	1.3350	1.3278	1.3209	1.3143	1.3079	1.3018	1.2959	1.2902	1.2847	1.2794	1.2742	1.2691	1.2642
240	1.3718	1.3675	1.3591	1.3512	1.3436	1.3365	1.3296	1.3231	1.3168	1.3107	1.3049	1.2993	1.2938	1.2885	1.2834	1.2784	1.2736
250	1.3801	1.3758	1.3675	1.3596	1.3521	1.3450	1.3382	1.3317	1.3255	1.3195	1 3137	1.3081	1.3027	1.2975	1.2924	1.2875	1.2827
260	1.3883	1.3840	1.3757	1.3679	1.3605	1.3534	1.3467	1.3402	1.3340	1.3281	1.3224	1.3168	1.3115	1.3063	1.3013	1.2964	1.2917
270	1.3964	1.3921	1.3838	1.3761	1.3687	1.3617	1.3550	1.3486	1.3424	1.3365	1.3308	1.3253	1.3200	1.3149	1.3099	1.3051	1.3004
280	1.4043	1.4001	1.3919	1.3841	1.3786	1.3698	1.3631	1.3568	1.3507	1 3448	1.3392	1.3337	1.3285	1.3234	1.3184	1.3137	1.3090
290	1.4122	1.4079	1.3998	1.3921	1.3847	1.3778	1.3712	1.3649	1.3588	1 3530	1.3474	1.3420	1.3367	1.3317	1.3268	1.3221	1.3175
300	1.4199	1.4157	1.4076	1.3999	1.3926	1.3857	1.3791	1.3728	1.3668	1.3610	1.3554	1.3501	1.3449	1.3399	1.3350	1.3303	1.3257
320	1.4352	1.4310	1.4229	1.4153	1.4081	1.4012	1.3947	1.3884	1.3825	1.3767	1 3712	1.3659	1.3608	1.3559	1.3511	1.3464	1.3419
340	—	1 4459	1.4379	1.4303	1.4231	1.4163	1.4099	1.4037	1.3978	1.3921	1.3866	1.3814	1.3763	1.3714	1.3667	1.3621	1.3576
360	—	—	—	—	—	—	1.4247	1.4186	1.4127	1 4070	1.4016	1 3964	1.3914	1.3866	1.3819	1.3773	1.3729
380	—	—	—	—	—	—	1 4392	1.4331	1.4273	1.4217	1. 163	1 4111	1.4062	1.4014	1.3967	1.3922	1.3878
400	—	—	—	—	—	—	—	—	—	—	—	1 4255	1.4206	1.4158	1.4112	1.4067	1.4024

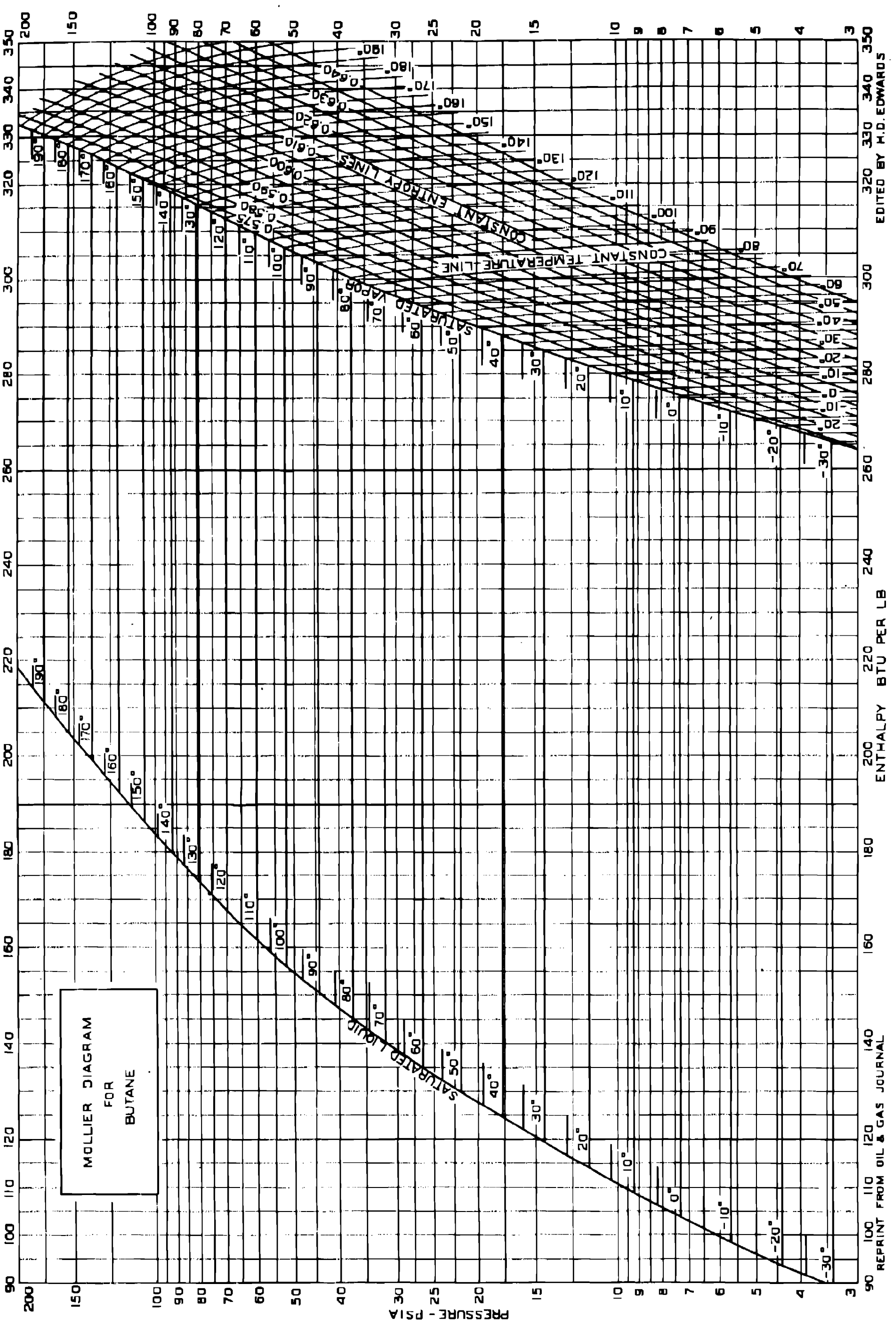
MOLLIER DIAGRAM FOR BUTANE
ENTHALPY BTU PER LB
PRESSURE - PSIA
EDITED BY H.D. EDWARDS
REPRINT FROM OIL & GAS JOURNAL
SATURATED LIQUID
SATURATED VAPORS
CONSTANT ENTROPY LINES
CONSTANT TEMPERATURE LINE
90
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Chart 3

Table 5. Butane (Normal)—Properties of Liquid and Saturated Vapor

(Enthalpies and entropies are referred to saturated liquid at −200 F where the values are zero. Refer to Chart 2 for enthalpies and entropies of superheated butane vapor.)

Temp F t	Pressure†		Liquid density†	Vapor, sp vol	Enthalpy** Btu per lb		Entropy** Btu per lb F	
	psia	psig	lb per cu ft $1/v_f$	cu ft per lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
0	7.3	*15.0	38.59	11.1	103.8	275.3	0.226	0.572
5	8.2	*13.2	38.41	9.98	106.2	276.9	0.231	0.572
10	9.2	*11.1	38.24	8.95	108.8	278.5	0.236	0.574
15	10.4	*8.8	38.07	8.05	111.5	280.0	0.242	0.571
20	11.6	*6.3	37.89	7.23	114.0	281.6	0.248	0.571
25	13.0	*3.6	37.72	6.55	116.7	283.1	0.254	0.571
30	14.4	*0.6	37.54	5.90	119.2	284.9	0.260	0.571
35	16.0	1.3	37.37	5.37	121.8	286.4	0.264	0.571
40	17.7	3.0	37.19	4.77	124.2	288.0	0.270	0.571
45	19.6	4.9	37.00	4.47	126.8	289.5	0.276	0.571
50	21.6	6.9	36.82	4.07	129.6	291.2	0.282	0.571
55	23.8	9.1	36.63	3.73	132.1	292.7	0.287	0.570
60	26.3	11.6	36.45	3.40	134.8	294.2	0.293	0.570
65	28.9	14.2	36.24	3.12	137.3	295.9	0.298	0.570
70	31.6	16.9	36.06	2.98	140.1	297.5	0.304	0.570
75	34.5	19.8	35.86	2.65	142.6	298.9	0.310	0.570
80	37.6	22.9	35.65	2.46	145.0	300.3	0.315	0.570
85	40.9	26.2	35.45	2.28	147.8	302.0	0.322	0.570
90	44.5	29.8	35.24	2.10	150.5	303.5	0.326	0.571
95	48.2	33.5	35.04	1.96	153.1	305.0	0.332	0.571
100	52.2	37.5	34.84	1.81	156.2	306.7	0.340	0.571
105	56.4	41.7	34.62	1.70	159.1	308.1	0.346	0.572
110	60.8	46.1	34.41	1.58	161.9	309.5	0.352	0.572
115	65.6	50.9	34.19	1.48	165.0	311.1	0.359	0.572
120	70.8	56.1	33.96	1.37	167.8	312.7	0.365	0.572
125	76.0	61.3	33.77	1.28	171.0	314.0	0.372	0.573
130	81.4	66.7	33.56	1.19	174.0	315.5	0.378	0.573
135	87.0	72.3	33.34	1.11	177.1	317.0	0.385	0.573
140	92.6	77.9	33.14	1.04	179.9	318.2	0.391	0.574
145	100.0	85.3	32.92	0.966	183.1	319.5	0.398	0.574
150	108.0	93.7	32.70	0.897	186.5	321.0	0.405	0.574
155	115.0	100.3	32.43	0.840	189.3	322.3	0.411	0.574
160	122.0	107.3	32.15	0.785	192.7	323.8	0.418	0.575
165	130.0	115.3	31.90	0.733	195.8	325.0	0.425	0.575
170	140.0	125.3	31.62	0.687	199.2	326.1	0.433	0.575
175	150.0	135.3	31.36	0.643	202.1	327.2	0.439	0.575
180	160.0	145.3	31.10	0.602	205.0	328.4	0.445	0.575

† Based on material from Dana Jenkins, Burdick and Timm, published originally in *Refrigerating Engineering*, June, 1926, vol. 12, no. 12, page 402.

* Inches of mercury below one standard atmosphere (29.92 in.).

** From Mollier Diagrams for Butane, W. C. Edmister, Standard Oil Co. (Indiana).

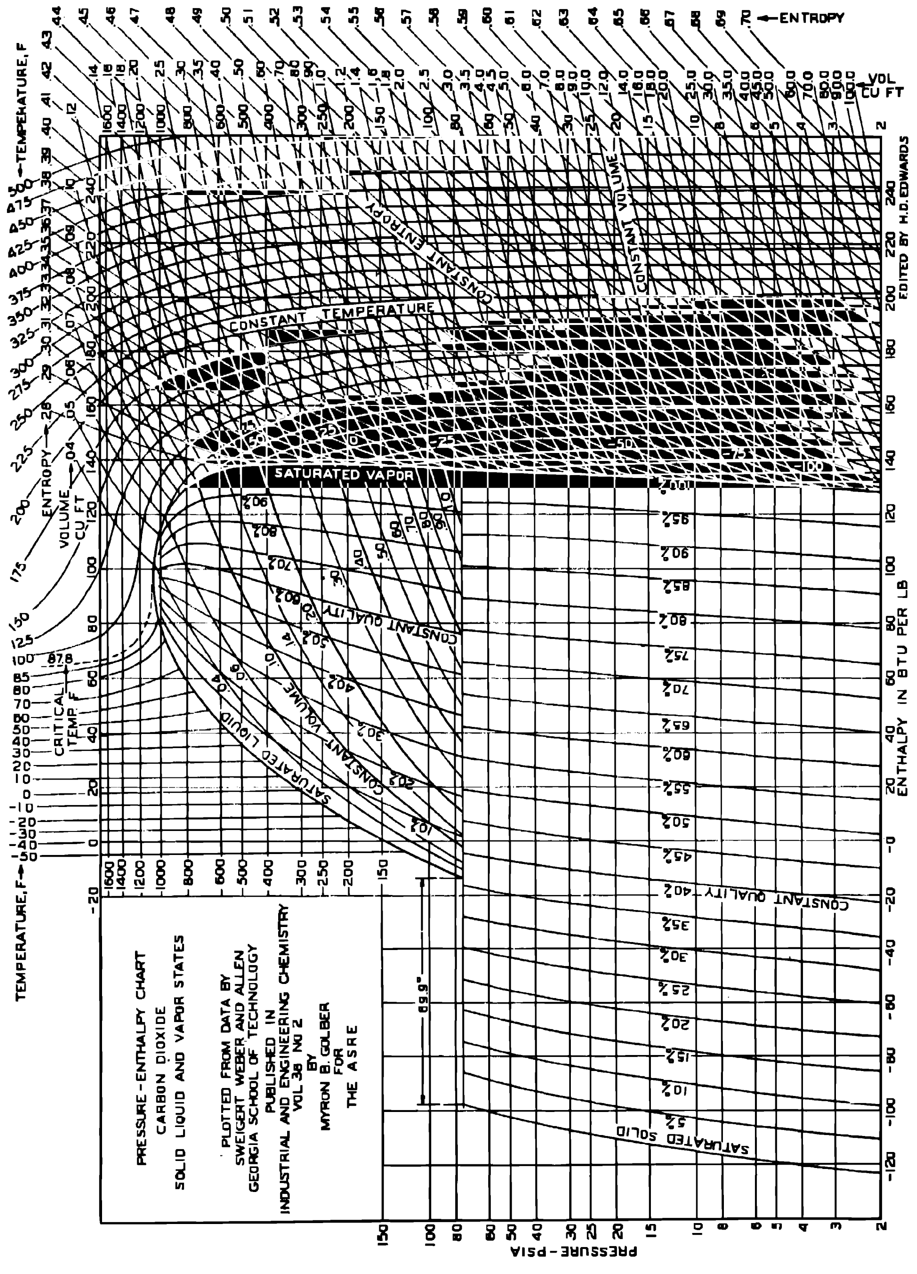

PRESSURE-ENTHALPY CHART
CARBON DIOXIDE
SOLID LIQUID AND VAPOR STATES
PLOTTED FROM DATA BY
SWEIGERT WEBER AND ALLEN
GEORGIA SCHOOL OF TECHNOLOGY
PUBLISHED IN
INDUSTRIAL AND ENGINEERING CHEMISTRY
VOL 38 NO 2
BY
MYRON B. GOLBER
FOR
THE ASRE
EDITED BY H.D. EDWARDS
ENTHALPY IN BTU PER LB
PRESSURE - PSIA
TEMPERATURE, F
ENTROPY
VOL CU FT
VOLUME CU FT
CRITICAL TEMP F
87.8
69.9"
CONSTANT TEMPERATURE
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT QUALITY
SATURATED VAPOR
SATURATED LIQUID
SATURATED SOLID

Chart 4

Table 6. Carbon Dioxide—Properties of Solid, Liquid and Saturated Vapor

	Temp F t	Pressure		Specific Volume cu ft per lb		Enthalpy, datum -40 F Btu per lb		Entropy, datum -40 F Btu per lb F	
		psia	psig	Solid or liquid v_i or v_f	Vapor v_g	Solid or liquid h_i or h_f	Vapor h_g	Solid or liquid s_f	Vapor s_g
Solid and Vapor	-147	2.14	25.56*	0.01004	35.80	-123.3	128.2	-.3214	+.4832
	-145	2.43	24.97*	0.01005	32.40	-122.8	128.5	-.1396	.4792
	-140	3.19	23.41*	0.01007	24.50	-121.4	129.3	-.3153	.4691
	-135	4.16	21.43*	0.01009	18.70	-120.1	130.0	-.3110	.4593
	-130	5.39	18.92*	0.01012	14.74	-118.7	130.7	-.3068	.4500
	-125	6.98	15.68*	0.01015	11.56	-117.4	131.3	-.3027	.4409
	-120	8.85	11.87*	0.01018	9.13	-116.0	132.1	-.2986	.4318
	-115	11.20	7.08*	0.01022	7.27	-114.3	132.7	-.2944	.4230
	-110	14.22	0.92*	0.01024	5.85	-113.1	133.3	-.2904	.4145
	-109.4	14.67	0.00	0.01025	5.69	-112.9	133.4	-.2898	.4134
	-105	17.80	3.13	0.01028	4.72	-111.5	133.9	-.2860	.4062
	-100	22.34	7.67	0.01032	3.80	-110.0	134.4	-.2815	.3981
	- 95	27.63	12.96	0.01036	3.09	-108.3	134.9	-.2768	.3902
	- 90	34.05	19.38	0.01040	2.52	-106.5	135.3	-.2720	.3822
	- 85	41.67	27.00	0.01044	2.07	-104.5	135.6	-.2667	.3742
	- 80	50.70	36.03	0.01049	1.70	-102.3	135.8	-.2610	.3665
	- 79	52.80	38.13	0.01050	1.63	-101.9	135.8	-.2599	.3649
	- 78	55.00	40.33	0.01051	1.56	-101.4	135.9	-.2587	.3633
	- 77	57.20	42.53	0.01052	1.51	-101.0	135.9	-.2575	.3617
	- 76	59.44	44.77	0.01053	1.46	-100.5	135.9	-.2563	.3601
	- 75	61.75	47.08	0.01054	1.40	-100.1	135.9	-.2551	.3585
	- 74	64.25	49.58	0.01055	1.35	- 99.7	135.9	-.2539	.3570
	- 73	66.75	52.08	0.01056	1.30	- 99.2	135.9	-.2528	.3554
	- 72	69.40	54.73	0.01057	1.25	- 98.8	136.0	-.2516	.3539
	- 71	72.10	57.43	0.01058	1.21	- 98.4	136.0	-.2505	.3523
	- 70	74.90	60.23	0.01059	1.17	- 98.0	136.0	-.2494	.3508
	- 69.9	75.1	60.4	0.01059	1.16	- 97.8	136.0	-.2493	.3506
				Triple Point					
Liquid and Vapor	- 69.9	75.1	60.4	0.01360	1.1570	- 13.7	136.0	-.0333	.3506
	- 68	78.59	63.92	0.01363	1.1095	- 12.8	136.2	-.0312	.3491
	- 66	82.42	67.75	0.01369	1.0590	- 11.9	136.3	-.0290	.3475
	- 64	86.39	71.72	0.01373	1.0100	- 10.9	136.4	-.0266	.3460
	- 62	90.49	75.82	0.01378	0.9650	- 10.1	136.6	-.0243	.3444
	- 60	94.75	80.08	0.01384	0.9520	- 9.1	136.7	-.0221	.3429
	- 58	99.15	84.48	0.01389	0.8875	- 8.2	136.8	-.0198	.3413
	- 56	103.69	89.02	0.01393	0.8520	- 7.3	137.0	-.0175	.3398
	- 54	108.40	93.73	0.01398	0.8180	- 6.4	137.1	-.0153	.3383
	- 52	113.25	98.58	0.01403	0.7840	- 5.5	137.2	-.0131	.3368
	- 50	118.27	103.60	0.01409	0.7500	- 4.6	137.3	-.0109	.3354
	- 48	123.45	108.78	0.01414	0.7200	- 3.6	137.5	-.0087	.3339
	- 46	128.80	114.13	0.01419	0.6930	- 2.7	137.6	-.0065	.3325
	- 44	134.31	119.64	0.01425	0.6660	- 1.8	137.7	-.0043	.3311
	- 42	140.00	125.33	0.01430	0.6380	- 0.9	137.8	-.0021	.3297
	- 40	145.87	131.20	0.01437	0.6113	0.0	137.9	.0000	.3285
	- 38	151.92	137.25	0.01142	0.5880	+ 1.0	138.0	+.0022	.3271
	- 36	158.15	143.48	0.01447	0.5650	1.9	138.1	.0043	.3258
	- 34	164.56	149.89	0.01454	0.5420	2.8	138.2	.0065	.3245
	- 32	171.17	156.50	0.01458	0.5210	3.8	138.3	.0085	.3232
	- 30	177.97	163.30	0.01465	0.5025	4.7	138.3	.0106	.3219
	- 28	184.97	170.30	0.01472	0.4845	5.6	138.4	.0126	.3205
	- 26	192.17	177.50	0.01478	0.4670	6.5	138.5	.0147	.3193
	- 24	199.57	184.90	0.01485	0.4500	7.4	138.6	.0168	.3180
	- 22	207.19	192.58	0.01491	0.4325	8.3	138.6	.0190	.3167
	- 20	215.02	200.35	0.01498	0.4165	9.2	138.7	.0210	.3155
	- 18	223.06	208.39	0.01504	0.4015	10.3	138.7	.0231	.3142
	- 16	231.32	216.65	0.01511	0.3865	11.2	138.8	.0252	.3130
	- 14	239.81	225.14	0.01518	0.3725	12.1	138.8	.0272	.3117
	- 12	248.52	233.85	0.01525	0.3590	12.9	138.8	.0293	.3104
	- 10	257.46	242.79	0.01533	0.3465	13.9	138.9	.0314	.3091
	- 8	266.63	251.96	0.01540	.3345	15.0	138.9	.0335	.3079
	- 6	276.05	261.38	0.01547	.3228	15.9	138.9	.0356	.3067
	- 4	285.70	271.03	0.01555	.3118	16.9	138.9	.0376	.3054
	- 2	295.61	280.94	0.01563	.3012	17.9	138.9	.0397	.3042
	0	305.76	291.09	0.01571	0.2905	18.8	138.9	.0419	.3030

* Inches of mercury below one atmosphere.

Table 6. Carbon Dioxide—Properties of Solid, Liquid and Saturated Vapor (Concluded)

	Temp F t	Pressure		Specific Volume cu ft per lb		Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
		psia	psig	Solid or liquid v_i or v_f	Vapor v_g	Solid or liquid h_i or h_f	Vapor h_g	Solid or liquid s_f	Vapor s_g
Liquid and Vapor	+ 2	316.2	301.5	0.01579	0.2800	19.8	138.9	.0441	.3018
	4	326.8	312.1	0.01587	.2708	20.8	138.9	.0462	.3006
	6	337.7	323.0	0.01597	.2610	21.8	138.9	.0483	.2994
	8	348.9	334.2	0.01605	.2520	22.9	138.8	.0503	.2982
	10	360.4	345.7	0.01614	.2435	24.0	138.8	.0525	.2970
	12	372.1	357.5	0.01623	0.2350	25.0	138.7	.0547	.2958
	14	384.1	369.4	0.01632	.2274	26.2	138.7	.0569	.2945
	16	396.4	381.7	0.01642	2195	27.3	138.6	.0591	.2933
	18	409.0	394.3	0.01650	.2120	28.5	138.6	.0613	.2921
	20	421.8	407.1	0.01662	.2048	29.6	138.5	.0636	2909
	22	435.0	420.3	0.01672	0.1978	30.8	138.4	.0660	.2897
	24	448.4	433.7	0.01684	.1910	31.8	138.3	.0684	.2885
	26	462.2	447.5	0.01695	.1845	33.1	138.1	.0707	.2873
	28	476.3	461.6	0.01707	.1782	34.3	138.0	.0730	.2861
	30	490.6	475.9	0.01719	.1720	35.6	137.8	.0753	.2849
	32	505.3	490.6	0.01731	0.1663	36.8	137.7	.0778	.2837
	34	520.3	505.6	0.01743	.1605	38.0	137.5	.0800	.2883
	36	535.7	521.0	0.01757	.1550	39.3	137.3	.0823	.2807
	38	551.3	536.6	0.01771	.1495	40.5	137.0	.0848	.2791
	40	567.3	552.6	0.01786	.1442	41.8	136.8	.0872	.2775
	42	583.7	569.0	0.01801	.1390	43.1	136.5	.0898	.2759
	44	600.4	585.7	0.01817	.1342	44.4	136.2	.0922	.2745
	46	617.5	602.8	0.01834	.1298	45.7	135.8	.0947	.2730
	48	634.9	620.2	0.01842	.1250	47.1	135.5	.0972	.2715
	50	652.7	638.0	0.01867	.1204	48.5	135.1	.1000	.2699
	52	670.8	656.1	0.01887	0.1163	49.9	134.6	.1026	.2681
	54	689.4	674.7	0.01906	.1122	51.3	134.0	.1052	.2664
	56	708.3	693.6	0.01926	.1080	52.8	133.4	.1080	.2645
	58	727.6	712.9	0.01948	.1038	54.2	132.8	.1108	.2627
	60	747.4	732.7	0.01970	.0995	55.7	132.2	.1136	.2606
	62	767.5	752.8	0.01997	0.0960	57.2	131.5	.1164	.2585
	64	788.1	773.4	0.02012	.0920	58.8	130.7	.1192	.2563
	66	809.1	794.4	0.02049	.0880	60.4	129.8	.1221	.2540
	68	830.6	815.9	0.02079	.0842	62.0	128.9	.1250	.2516
	70	852.5	837.8	0.02109	.0800	63.7	127.8	.1282	.2485
	72	874.9	860.2	0.02146	0.0760	65.6	126.5	.1316	.2455
	74	897.7	883.0	0.02188	.0720	67.5	124.7	.1351	.2423
	76	921.1	906.4	0.02240	.0680	69.6	122.9	.1390	.2385
	78	945.0	930.3	0.02300	.0640	71.8	121.1	.1430	.2347
	80	969.3	954.6	0.02370	.0600	74.0	119.0	.1469	.2305
	82	994.2	979.5	0.02458	0.0560	76.6	116.6	.1514	.2258
	84	1019.7	1005.0	0.02556	.0520	79.6	114.0	.1564	.2205
	86	1045.7	1031.0	0.02686	.0479	83.4	110.5	.1636	.2133
	87.8	1072.1	1057.4	0.03453	.0345	97.1	97.1	.1880	.1880

Table 7. Carbon Dioxide—Specific Volume of Superheated Vapor (cu ft per lb)

Pressure psia	Temperature, F											
	−75	−50	0	50	100	150	200	250	300	350	400	450
.20	469.4	499.9	560.9	621.9	682.8	743.8	804.8	865.8	926.7	987.7	—	—
.30	312.9	333.3	373.9	414.6	455.2	495.9	536.5	577.2	617.8	658.5	699.1	739.7
.40	234.7	249.9	280.4	310.9	341.4	371.9	402.4	432.9	463.3	493.8	524.3	554.8
.50	187.7	199.9	224.3	248.7	273.1	297.5	321.9	346.3	370.7	395.0	419.4	443.8
.60	156.4	166.6	186.9	207.3	227.6	247.9	268.2	288.6	308.9	329.2	349.5	369.8
.70	134.1	141.9	160.2	177.6	195.1	212.5	229.9	247.3	264.7	282.2	299.6	317.0
.80	117.4	125.0	140.2	155.5	170.7	186.0	201.2	216.5	231.7	246.9	262.2	277.4
.90	104.3	111.1	124.7	138.2	151.8	165.3	178.9	192.4	206.0	219.5	233.0	246.5
1.00	93.90	100.0	112.2	124.4	136.6	148.8	161.0	173.2	185.4	197.5	209.7	221.9
2.0	46.82	49.96	56.03	62.12	68.29	74.39	80.48	86.58	92.67	98.77	104.8	110.09
3.0	31.18	33.28	37.32	41.40	45.49	49.55	53.64	57.72	61.78	65.85	69.91	73.97
4.0	23.38	24.93	27.98	31.03	34.09	37.13	40.22	43.29	46.33	49.38	52.43	55.48
5.0	18.66	19.91	22.37	24.82	27.26	29.72	32.17	34.61	39.06	39.51	41.93	44.38
6.0	15.53	16.56	18.63	20.67	22.71	24.76	26.80	28.84	30.87	32.92	34.94	36.98
8.0	11.62	12.40	13.96	15.49	17.02	18.56	20.09	21.62	23.14	24.60	26.20	27.74
10.0	9.280	9.902	11.15	12.38	13.61	14.84	16.06	17.29	18.51	19.74	20.96	22.19
12.0	7.715	8.236	9.280	10.31	11.33	12.37	13.38	14.41	15.42	16.44	17.46	18.49
16.0	5.769	6.177	6.944	7.716	8.485	9.266	10.03	10.80	11.56	12.33	13.09	13.86
20.0	4.586	4.904	5.542	6.119	6.778	7.407	8.016	8.633	9.247	9.861	10.47	11.09
24.0	3.803	4.070	4.607	5.125	5.640	6.167	6.674	7.190	7.701	8.215	8.724	9.236
28.0	3.246	3.471	3.939	4.385	4.826	5.282	5.718	6.160	6.599	7.039	7.475	7.918
32.0	2.825	3.028	3.439	3.829	4.217	4.617	4.999	5.387	5.773	6.157	6.541	6.925
36.0	2.499	2.681	3.050	3.398	3.743	4.101	4.441	4.785	5.130	5.470	5.812	6.156
40.0	2.239	2.404	2.738	3.053	3.363	3.688	3.993	4.304	4.615	4.922	5.230	5.538
44.0	2.025	2.177	2.483	2.771	3.054	3.349	3.628	3.972	4.193	4.473	4.753	5.034
48.0	1.847	1.987	2.270	2.535	2.795	3.068	3.323	3.587	3.843	4.099	4.355	4.614
52.0	1.696	1.827	2.091	2.335	2.576	2.829	3.065	3.306	3.545	3.783	4.019	4.258
60.0	1.456	1.571	1.803	2.016	2.225	2.448	2.652	2.862	3.070	3.276	3.481	3.689
70.0	—	1.329	1.536	1.720	1.901	2.094	2.269	2.450	2.628	2.806	2.982	3.071
80.0	—	1.154	1.335	1.498	1.657	1.828	1.982	2.141	2.298	2.453	2.608	2.764
90.0	—	1.015	1.179	1.325	1.476	1.621	1.758	1.900	2.040	2.178	2.317	2.457
100.0	—	0.905	1.055	1.187	1.316	1.456	1.579	1.708	1.834	1.960	2.083	2.210
120.0	—	—	0.8655	0.9799	1.088	1.208	1.311	1.420	1.525	1.630	1.734	1.840
160.0	—	—	0.6305	0.7207	0.8033	0.8986	0.9760	1.059	1.139	1.219	1.297	1.378
200.0	—	—	0.4891	0.5652	0.6376	0.7125	0.7748	0.8424	0.9075	0.9720	1.035	1.100
240.0	—	—	0.3948	0.4614	0.5237	0.5886	0.6407	0.6981	0.7532	0.8074	0.8604	0.9152
280.0	—	—	0.3286	0.3865	0.4425	0.4994	0.5448	0.5950	0.6428	0.6896	0.7358	0.7833
320.0	—	—	—	0.3298	0.3815	0.4322	0.4729	0.5177	0.5601	0.6015	0.6422	0.6840
360.0	—	—	—	0.2858	0.3341	0.3780	0.4171	0.4576	0.4958	0.5329	0.5693	0.6071
400.0	—	—	—	0.2505	0.2961	0.3383	0.3724	0.4095	0.4443	0.4781	0.5110	0.5455
440.0	—	—	—	0.2216	0.2652	0.3040	0.3358	0.3701	0.4022	0.4332	0.4633	0.4950
520.0	—	—	—	0.1772	0.2174	0.2513	0.2795	0.3096	0.3374	0.3641	0.3901	0.4174
560.0	—	—	—	0.1598	0.1986	0.2304	0.2574	0.2858	0.3119	0.3369	0.3613	0.3869
600.0	—	—	—	0.1452	0.1823	0.2123	0.2383	0.2652	0.2898	0.3135	0.3363	0.3605
650.0	—	—	—	—	0.1648	0.1928	0.2176	0.2430	0.2661	0.2881	0.3094	0.3323
700.0	—	—	—	—	0.1453	0.1760	0.2000	0.2239	0.2457	0.2664	0.2864	0.3075
750.0	—	—	—	—	0.1320	0.1615	0.1846	0.2074	0.2281	0.2476	0.2664	0.2865
800.0	—	—	—	—	0.1196	0.1483	0.1712	0.1930	0.2126	0.2311	0.2489	0.2681
850.0	—	—	—	—	0.1089	0.1373	0.1594	0.1803	0.1990	0.2166	0.2335	0.2518
900.0	—	—	—	—	0.1000	0.1273	0.1489	0.1690	0.1869	0.2037	0.2198	0.2372
950.0	—	—	—	—	—	0.1183	0.1394	0.1588	0.1761	0.1921	0.2075	0.2243
1000.0	—	—	—	—	—	0.1101	0.1310	0.1497	0.1663	0.1819	0.1966	0.2126
1100.0	—	—	—	—	—	—	0.1164	0.1339	0.1495	0.1646	0.1778	0.1925
1200.0	—	—	—	—	—	—	0.1042	0.1208	0.1356	0.1493	0.1621	0.1756
1300.0	—	—	—	—	—	—	—	0.1100	0.1237	0.1368	0.1488	0.1614
1400.0	—	—	—	—	—	—	—	—	0.1136	0.1260	0.1375	0.1492
1500.0	—	—	—	—	—	—	—	—	0.1049	0.1167	0.1277	0.1386
1600.0	—	—	—	—	—	—	—	—	—	0.1086	0.1191	0.1294
1700.0	—	—	—	—	—	—	—	—	—	0.1014	0.1155	0.1213
1800.0	—	—	—	—	—	—	—	—	—	—	0.1047	0.1140
1900.0	—	—	—	—	—	—	—	—	—	—	—	0.1075
2000.0	—	—	—	—	—	—	—	—	—	—	—	0.1017

Table 8. Carbon Dioxide—Enthalpy of Superheated Vapor (Btu per lb from −40 F)

Pressure psia	Temperature, F											
	−75	−50	0	50	100	150	200	250	300	350	400	450
0.20	140.0	144.8	154.6	164.5	174.8	185.2	195.9	206.9	218.2	229.7	—	—
2.0	140.0	144.8	154.6	164.5	174.8	185.2	195.9	206.9	218.2	229.7	241.5	253.6
3.0	139.9	144.7	154.5	164.4	174.8	185.2	195.9	206.9	218.2	229.7	241.5	253.6
4.0	139.8	144.7	154.5	164.4	174.7	185.2	195.9	206.9	218.2	229.7	241.5	253.6
5.0	139.8	144.6	154.4	164.4	174.7	185.1	195.9	206.9	218.2	229.7	241.5	253.6
6.0	139.7	144.5	154.4	164.3	174.6	185.1	195.9	206.9	218.2	229.7	241.5	253.6
8.0	139.6	144.5	154.3	164.2	174.6	185.1	195.8	206.8	218.1	229.6	241.5	253.6
10.0	139.4	144.3	154.1	164.1	174.5	185.0	195.8	206.8	218.1	229.6	241.4	253.6
12.0	139.3	144.2	154.0	164.0	174.4	185.0	195.7	206.7	218.1	229.6	241.4	253.5
16.0	139.0	144.0	153.8	163.8	174.2	184.8	195.7	206.7	218.0	229.5	241.4	253.5
20.0	138.7	143.8	153.6	163.6	174.1	184.7	195.6	206.6	217.9	229.5	241.3	253.5
24.0	138.5	143.6	153.4	163.4	173.9	184.6	195.5	206.5	217.9	229.4	241.3	253.4
28.0	138.2	143.4	153.2	163.2	173.8	184.5	195.4	206.5	217.8	229.4	241.3	253.4
32.0	137.9	143.2	153.0	163.1	173.6	184.4	195.4	206.4	217.8	229.3	241.2	253.4
36.0	137.7	142.9	152.8	162.9	173.5	184.3	195.3	206.3	217.7	229.3	241.2	253.3
40.0	137.4	142.7	152.6	162.7	173.3	184.2	195.2	206.3	217.7	229.3	241.1	253.3
44.0	137.2	142.5	152.4	162.5	173.2	184.1	195.2	206.2	217.6	229.2	241.1	253.3
48.0	136.9	142.3	152.2	162.3	173.0	184.0	195.1	206.1	217.5	229.1	241.0	253.2
52.0	136.6	142.1	152.0	162.2	172.9	183.9	195.0	206.1	217.5	229.1	241.0	253.2
56.0	136.4	141.9	151.8	162.0	172.8	183.8	195.0	206.0	217.4	229.1	241.0	253.2
60.0	136.1	141.7	151.6	161.8	172.6	183.7	194.9	205.9	217.3	228.9	240.9	253.1
70.0	—	141.2	151.1	161.3	172.2	183.5	194.7	205.7	217.2	228.8	240.8	253.0
80.0		140.6	150.6	160.9	171.9	183.2	194.5	205.6	217.0	228.7	240.7	252.9
90.0	—	140.1	150.1	160.4	171.5	183.0	194.3	205.4	216.9	228.6	240.6	252.8
100.0	—	139.6	149.5	160.0	171.1	182.7	194.1	205.2	216.8	228.5	240.5	252.7
120.0	—	—	148.5	159.0	170.4	182.2	193.8	204.9	216.5	228.3	240.3	252.6
140.0		—	147.5	158.1	169.6	181.7	193.5	204.6	216.2	228.0	240.1	252.4
160.0	—	—	146.5	157.2	168.9	181.2	193.1	204.2	215.9	227.8	239.9	252.2
180.0			145.5	156.3	168.1	180.7	192.8	203.9	215.6	227.5	239.7	252.1
200.0			144.5	155.4	167.4	180.2	192.4	203.5	215.3	227.3	239.5	252.0
220.0			143.4	154.5	166.6	179.7	192.1	203.2	215.0	227.0	239.3	251.8
240.0		—	142.4	153.5	165.9	179.2	191.7	202.9	214.8	226.8	239.1	251.6
260.0		—	141.4	152.6	165.2	178.7	191.4	202.5	214.5	226.6	238.9	251.5
280.0			140.4	151.7	164.4	178.2	191.0	202.2	214.2	226.3	238.6	251.3
300.0			—	150.8	163.7	177.7	190.7	201.9	213.9	226.1	238.4	251.1
320.0			—	149.9	162.9	177.2	190.3	201.5	213.6	225.8	238.2	251.0
340.0			—	149.0	162.2	176.7	190.0	201.2	213.3	225.6	238.0	250.8
360.0			—	148.0	161.4	176.2	189.6	200.8	213.1	225.4	237.8	250.5
380.0				147.1	160.7	175.7	189.3	200.5	212.8	225.1	237.6	250.3
400.0			—	146.2	159.9	175.7	188.9	200.2	212.5	224.9	237.4	250.2
420.0				145.3	159.2	174.7	188.6	199.8	212.2	224.6	237.2	250.0
440.0				144.4	158.4	174.2	188.2	199.5	211.9	224.4	237.0	249.8
480.0				142.5	156.9	173.2	187.5	198.8	211.3	223.9	236.6	249.5
520.0				140.7	155.5	172.2	186.8	198.2	210.8	223.4	236.2	249.1
560.0				138.9	154.0	171.2	186.1	197.5	210.2	223.0	235.8	248.8
600.0				137.1	152.5	170.2	185.4	196.8	209.6	222.5	235.4	248.5
700.0			—	—	148.8	167.7	183.7	195.1	208.2	221.3	234.4	247.6
800.0				—	145.0	165.2	181.9	193.4	206.8	220.1	233.3	246.7
900.0			—	—	141.3	162.7	180.2	191.7	205.3	218.9	232.3	245.9
1000.0	—			—	—	160.2	178.4	190.1	203.9	217.7	231.3	245.0
1100.0					—	—	177.5	188.4	202.5	216.5	230.3	244.1
1200.0	—		—		—	—	175.2	186.7	201.0	215.3	229.3	243.3
1300.0	—		—		—	—	—	185.0	199.6	214.1	228.2	242.4
1400.0	—	—	—		—	—	—	—	198.2	212.9	227.2	241.6
1500.0	—	—	—	—	—		—	—	196.7	211.7	225.9	240.7
1600.0	—	—	—	—	—	—	—	—	—	210.5	224.4	239.8
1700.0	—	—	—	—	—	—	—	—	—	209.0	222.8	239.0
1800.0	—	—	—	—	—	—	—	—	—	—	220.8	238.1
1900.0	—	—	—	—	—	—	—	—	—	—	—	237.3
2000.0	—	—	—	—	—	—	—	—	—	—	—	236.2

Table 9. Carbon Dioxide—Entropy of Superheated Vapor (Btu per lb F from −40 F)

Pressure psia	Temperature, F											
	−75	−50	0	50	100	150	200	250	300	350	400	450
.2	.6276	.6396	.6616	.6820	.7010	.7188	.7356	.7516	.7669	.7815	—	—
.3	.6093	.6213	.6433	.6637	.6827	.7005	.7173	.7333	.7486	.7632	.7772	.7909
.4	.5963	.6083	.6303	.6507	.6697	.6875	.7043	.7203	.7356	.7502	.7642	.7779
.6	.5780	.5900	.6120	.6324	.6512	.6692	.6860	.7020	.7173	.7319	.7459	.7596
.8	.5650	.5770	.5990	.6194	.6384	.6562	.6730	.6890	.7043	.7189	.7329	.7466
1.0	.5550	.5670	.5890	.6094	.6284	.6462	.6630	.6790	.6943	.7089	.7229	.7366
1.2	.5468	.5588	.5808	.6012	.6202	.6380	.6548	.6708	.6861	.7007	.7147	.7284
1.4	.5398	.5518	.5738	.5942	.6132	.6310	.6478	.6638	.6791	.6937	.7077	.7214
1.6	.5338	.5458	.5778	.5882	.6072	.6250	.6418	.6578	.6731	.6877	.7017	.7154
1.8	.5285	.5405	.5625	.5829	.6019	.6197	.6365	.6525	.6678	.6824	.6964	.7101
2.0	.5237	.5357	.5577	.5781	.5971	.6149	.6317	.6477	.6630	.6776	.6916	.7053
3.0	.5054	.5174	.5394	.5598	.5788	.5966	.6134	.6294	.6447	.6593	.6733	.6870
4.0	.4925	.5045	.5265	.5469	.5659	.5837	.6005	.6165	.6318	.6464	.6604	.6741
6.0	.4742	.4862	.5082	.5286	.5476	.5654	.5822	.5982	.6145	.6281	.6421	.6558
8.0	.4612	.4732	.4952	.5156	.5346	.5524	.5692	.5852	.6005	.6151	.6291	.6428
10.0	.4511	.4631	.4851	.5055	.5245	.5423	.5591	.5751	.5904	.6050	.6190	.6327
12.0	.4429	.4549	.4769	.4973	.5163	.5341	.5509	.5669	.5822	.5968	.6108	.6245
14.0	.4358	.4479	.4699	.4903	.5093	.5271	.5439	.5599	.5752	.5898	.6038	.6175
16.0	.4297	.4418	.4639	.4843	.5033	.5211	.5379	.5539	.5692	.5838	.5978	.6115
20.0	.4195	.4316	.4537	.4742	.4932	.5110	.5278	.5438	.5591	.5737	.5877	.6014
28.0	.4038	.4159	.4381	.4587	.4778	.4957	.5126	.5287	.5440	.5586	.5726	.5863
36.0	.3918	.4040	.4263	.4469	.4661	.4841	.5010	.5171	.5325	.5477	.5612	.5749
44.0	.3818	.3941	.4166	.4374	.4567	.4747	.4917	.5079	.5233	.5380	.5521	.5659
52.0	.3722	.3850	.4079	.4289	.4484	.4666	.4837	.4999	.5154	.5302	.5443	.5581
60.0	.3579	.3739	.3991	.4210	.4409	.4593	.4766	.4930	.5086	.5234	.5376	.5515
70.0	—	.3638	.3898	.4132	.4335	.4521	.4695	.4860	.5016	.5164	.5306	.5445
80.0	—	.3556	.3822	.4062	.4268	.4457	.4633	.4798	.4955	.5104	.5246	.5385
90.0	—	.3487	.3759	.4002	.4211	.4401	.4578	.4744	.4901	.5050	.5193	.5332
100.0	—	.3427	.3702	.3949	.4160	.4351	.4529	.4695	.4853	.5002	.5145	.5284
140.0	—	—	.3525	.3783	.3998	.4191	.4370	.4537	.4696	.4846	.4990	.5130
180.0	—	—	.3368	.3643	.3871	.4069	.4250	.4419	.4580	.4730	.4875	.5014
220.0	—	—	.3232	.3526	.3765	.3969	.4152	.4324	.4485	.4637	.4782	.4922
260.0	—	—	.3122	.3421	.3674	.3883	.4075	.4244	.4407	.4559	.4706	.4846
300.0	—	—	—	.3340	.3591	.3807	.3997	.4173	.4338	.4493	.4640	.4780
340.0	—	—	—	.3255	.3514	.3736	.3932	.4112	.4279	.4435	.4582	.4722
380.0	—	—	—	.3173	.3439	.3668	.3870	.4054	.4224	.4381	.4529	.4670
400.0	—	—	—	.3135	.3405	.3637	.3842	.4028	.4199	.4357	.4506	.4647
420.0	—	—	—	.3098	.3371	.3606	.3813	.4001	.4173	.4332	.4482	.4624
440.0	—	—	—	.3060	.3337	.3575	.3784	.3974	.4158	.4308	.4459	.4601
480.0	—	—	—	.2993	.3276	.3519	.3732	.3925	.4102	.4264	.4415	.4559
520.0	—	—	—	.2926	.3216	.3465	.3683	.3880	.4059	.4224	.4377	.4522
560.0	—	—	—	.2864	.3159	.3413	.3635	.3835	.4017	.4184	.4339	.4486
600.0	—	—	—	.2798	.3101	.3361	.3587	.3791	.3976	.4146	.4303	.4455
650.0	—	—	—	—	.3047	.3312	.3541	.3747	.3934	.4106	.4264	.4410
700.0	—	—	—	—	.2994	.3264	.3496	.3705	.3895	.4069	.4228	.4378
750.0	—	—	—	—	.2939	.3214	.3450	.3662	.3854	.4030	.4190	.4341
800.0	—	—	—	—	.2889	.3169	.3409	.3624	.3819	.3997	.4158	.4310
850.0	—	—	—	—	.2839	.3124	.3368	.3586	.3783	.3963	.4126	.4279
900.0	—	—	—	—	.2788	.3079	.3327	.3548	.3748	.3930	.4095	.4249
950.0	—	—	—	—	—	.3038	.3289	.3512	.3715	.3899	.4066	.4221
1000.0	—	—	—	—	—	.2996	.3250	.3475	.3681	.3867	.4036	.4193
1100.0	—	—	—	—	—	—	.3183	.3413	.3622	.3811	.3983	.4142
1200.0	—	—	—	—	—	—	.3121	.3357	.3569	.3761	.3936	.4097
1300.0	—	—	—	—	—	—	—	.3308	.3524	.3719	.3895	.4057
1400.0	—	—	—	—	—	—	—	—	.3481	.3679	.3856	.4020
1500.0	—	—	—	—	—	—	—	—	.3437	.3639	.3817	.3983
1600.0	—	—	—	—	—	—	—	—	—	.3600	.3780	.3948
1700.0	—	—	—	—	—	—	—	—	—	.3561	.3744	.3913
1800.0	—	—	—	—	—	—	—	—	—	—	.3708	.3879
1900.0	—	—	—	—	—	—	—	—	—	—	—	.3841
2000.0	—	—	—	—	—	—	—	—	—	—	—	.3807

Table 10. Carrene 7—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
40	10.84	7.86*	84.20	4.0764	0.00	89.92	0.00000	0.21424
38	11.42	6.68*	84.03	3.8825	0.51	90.19	.00122	.21388
-36	12.03	5.44*	83.86	3.7008	1.02	90.47	.00243	.21353
-34	12.66	4.15*	83.69	3.5279	1.54	90.75	.00365	.21319
-32	13.31	2.83*	83.51	3.3663	2.06	91.03	.00485	.21286
30	14.00	1.40*	83.34	3.2133	2.57	91.30	0.00606	0.21254
-28	14.71	0.01	83.16	3.0679	3.09	91.58	.00725	.21222
-26	15.45	0.75	82.99	2.9311	3.61	91.86	.00845	.21191
24	16.21	1.51	82.81	2.8015	4.13	92.13	.00963	.21159
-22	17.01	2.31	82.63	2.6789	4.66	92.41	.01083	.21131
-20	17.84	3.14	82.45	2.5627	5.18	92.68	0.01202	0.21102
-18	18.70	4.00	82.27	2.4525	5.70	92.95	.01322	.21074
-16	19.59	4.89	82.09	2.3482	6.23	93.23	.01438	.21045
14	20.51	5.81	81.91	2.2489	6.76	93.50	.01557	.21018
12	21.47	6.77	81.72	2.1552	7.28	93.77	.01674	.20991
10	22.46	7.76	81.54	2.0657	7.81	94.04	0.01793	0.20966
8	23.49	8.79	81.35	1.9811	8.34	94.31	.01909	.20940
6	24.55	9.85	81.16	1.9006	8.88	94.58	.02027	.20916
4	25.65	10.95	80.97	1.8242	9.41	94.85	.02144	.20892
2	26.79	12.09	80.78	1.7512	9.94	95.11	.02259	.20867
0	27.96	13.26	80.59	1.6817	10.48	95.38	0.02376	0.20844
2	29.18	14.48	80.40	1.6160	11.01	95.64	.02491	.20821
4	30.43	15.73	80.21	1.5532	11.56	95.91	.02608	.20799
6	31.73	17.03	80.01	1.4930	12.09	96.17	.02722	.20776
8	33.06	18.36	79.81	1.4361	12.63	96.43	.02839	.20755
10	34.45	19.75	79.62	1.3816	13.17	96.69	0.02954	0.20734
12	35.88	21.18	79.42	1.3295	13.71	96.95	.03067	.20712
14	37.35	22.65	79.22	1.2799	14.26	97.21	.03182	.20693
16	38.86	24.16	79.02	1.2326	14.81	97.47	.03296	.20672
18	40.42	25.72	78.81	1.1873	15.36	97.73	.03411	.20653
20	42.03	27.33	78.61	1.1441	15.91	97.98	0.03526	0.20635
22	43.69	28.99	78.41	1.1027	16.46	98.24	.03638	.20615
24	45.40	30.70	78.20	1.0632	17.01	98.49	.03752	.20597
26	47.15	32.45	77.99	1.0255	17.56	98.74	.03865	.20578
28	48.96	34.26	77.79	0.9892	18.12	98.99	.03978	.20560
30	50.82	36.12	77.58	0.9546	18.67	99.24	0.04092	0.20544
32	52.74	38.04	77.37	.9214	19.23	99.49	.04203	.20526
34	54.71	40.01	77.15	.8895	19.80	99.74	.04316	.20508
36	56.72	42.02	76.94	.8591	20.35	99.98	.04430	.20493
38	58.80	44.10	76.73	.8298	20.91	100.22	.04542	.20476
40	60.94	46.24	76.51	0.8018	21.47	100.46	0.04654	0.20460
42	63.14	48.44	76.30	.7748	22.03	100.70	.04764	.20443
44	65.39	50.69	76.08	.7490	22.60	100.94	.04875	.20427
46	67.71	53.01	75.86	.7242	23.17	101.18	.04986	.20411
48	70.09	55.39	75.64	.7002	23.75	101.41	.05099	.20395
50	72.52	57.82	75.42	0.6774	24.32	101.65	0.05212	0.20382
52	75.02	60.32	75.20	.6554	24.89	101.88	.05321	.20366
54	77.57	62.87	74.97	.6343	25.47	102.11	.05431	.20350
56	80.22	65.52	74.75	.6138	26.04	102.33	.05543	.20335
58	82.91	68.21	74.52	.5944	26.60	102.56	.05649	.20321
60	85.66	70.96	74.29	0.5756	27.19	102.78	0.05763	0.20307
62	88.49	73.79	74.07	.5575	27.76	103.00	.05872	.20293
64	91.39	76.69	73.84	.5401	28.33	103.22	.05979	.20278
66	94.37	79.67	73.61	.5232	28.93	103.44	.06091	.20264
68	97.41	82.71	73.37	.5069	29.52	103.65	.06203	.20250
70	100.51	85.81	73.14	0.4914	30.11	103.87	0.06311	0.20235
72	103.71	89.01	72.91	.4763	30.69	104.08	.06419	.20221
74	106.97	92.27	72.67	.4619	31.28	104.29	.06527	.20207
76	110.29	95.59	72.43	.4479	31.87	104.49	.06637	.20193
78	113.70	99.00	72.20	.4344	32.47	104.69	.06749	.20180
80	117.20	102.5	71.96	0.4213	33.06	104.89	0.06855	0.20164
82	120.78	106.1	71.72	.4088	33.63	105.09	.06959	.20150
84	124.40	109.7	71.48	.3967	34.25	105.29	.07071	.20136
86	128.14	113.4	71.23	.3849	34.85	105.48	.07179	.20122
88	131.96	117.3	70.99	.3736	35.45	105.67	.07288	.20109

* Inches of Mercury below one standard atmosphere (29.92 in.).

W. A. Pennington, Compiler.

Table 10. Carrene 7—Properties of Liquid and Saturated Vapor (Continued)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
90	135.86	121.2	70.74	0.3626	36.05	105.86	0.07395	0.20093
92	139.83	125.1	70.50	.3520	36.67	106.04	.07505	.20078
94	143.90	129.2	70.25	.3418	37.26	106.22	.07611	.20065
96	148.03	133.3	70.00	.3319	37.89	106.40	.07722	.20050
98	152.27	137.6	69.75	.3224	38.48	106.58	.07826	.20036
100	156.61	141.9	69.50	0.3131	39.10	106.75	0.07935	0.20021
102	161.02	146.3	69.25	.3041	39.71	106.91	.08042	.20005
104	165.55	150.9	69.00	.2954	40.34	107.08	.08151	.19990
106	170.14	155.4	68.74	.2870	40.97	107.24	.08261	.19975
108	174.84	160.1	68.49	.2788	41.59	107.39	.08367	.19958
110	179.62	164.9	68.23	0.2709	42.23	107.55	0.08479	0.19943
112	184.51	169.8	67.97	.2633	42.83	107.70	.08582	.19928
114	189.47	174.8	67.71	.2559	43.48	107.85	.08691	.19911
116	194.55	179.9	67.45	.2486	44.11	107.98	.08800	.19893
118	199.71	185.0	67.19	.2418	44.73	108.13	.08905	.19879
120	204.99	190.3	66.93	0.2350	45.36	108.26	0.09012	0.19862
122	210.40	195.7	66.66	.2283	46.03	108.38	.09124	.19842
124	215.88	201.2	66.40	.2219	46.69	108.50	.09236	.19825
126	221.44	206.7	66.13	.2157	47.33	108.62	.09342	.19806
128	227.13	212.4	65.86	.2097	47.99	108.74	.09451	.19787
130	232.89	218.2	65.60	0.2038	48.65	108.84	0.09560	0.19767
132	238.80	224.1	65.33	.1982	49.29	108.95	.09666	.19748
134	244.84	230.1	65.05	.1928	49.92	109.05	.09771	.19730
136	250.96	236.3	64.78	.1873	50.60	109.14	.09883	.19709
138	257.17	242.5	64.51	.1820	51.31	109.22	.09996	.19684
140	263.52	248.8	64.23	0.1768	52.02	109.29	0.10113	0.19662

W. A. Pennington, Compiler.

Table 11. Carrene 7—Properties of Superheated Vapor

Temp F	Pressure 10 psia Gage pressure 9.6 in. vac (Sat temp −43.0 F)			Pressure 12 psia Gage pressure 5.5 in. vac (Sat temp −36.1 F)			Pressure 14 psia Gage pressure 1.4 in. vac (Sat temp −30.0 F)			Pressure 16 psia Gage pressure 1.3 psig (Sat temp −24.6 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
−40	4.4276	89.96	0.21592	—	—	—	—	—	—	—	—	—
−30	4.5402	91.52	0.21959	3.7659	91.41	0.21578	—	—	—	—	—	—
−20	4.6525	93.09	0.22320	3.8602	92.99	0.21940	3.2938	92.88	0.21617	2.8691	92.78	0.21334
−10	4.7646	94.68	0.22677	3.9542	94.58	0.22298	3.3751	94.48	0.21975	2.9408	94.37	0.21693
0	4.8765	96.28	0.23029	4.0481	96.18	0.22651	3.4561	96.08	0.22328	3.0122	95.98	0.22047
10	4.9882	97.90	0.23377	4.1418	97.80	0.22999	3.5370	97.71	0.22677	3.0835	97.61	0.22397
20	5.0996	99.53	0.23721	4.2354	99.44	0.23343	3.6177	99.34	0.23022	3.1546	99.25	0.22742
30	5.2110	101.17	0.24060	4.3288	101.08	0.23684	3.6983	101.00	0.23363	3.2255	100.91	0.23084
40	5.3223	102.84	0.24396	4.4220	102.75	0.24020	3.7787	102.66	0.23700	3.2963	102.58	0.23421
50	5.4334	104.51	0.24729	4.5151	104.43	0.24353	3.8590	104.34	0.24033	3.3669	104.26	0.23755
60	5.5443	106.21	0.25057	4.6082	106.12	0.24682	3.9391	106.04	0.24363	3.4374	105.96	0.24085
70	5.6552	107.91	0.25383	4.7011	107.83	0.25008	4.0191	107.75	0.24689	3.5078	107.67	0.24412
80	5.7660	109.64	0.25705	4.7939	109.56	0.25331	4.0990	109.48	0.25012	3.5781	109.40	0.24735
90	5.8767	111.38	0.26024	4.8865	111.30	0.25650	4.1788	111.22	0.25332	3.6483	111.15	0.25055
100	5.9873	113.13	0.26340	4.9791	113.05	0.25966	4.2586	112.98	0.25649	3.7183	112.91	0.25373
110	6.0978	114.90	0.26653	5.0715	114.83	0.26280	4.3382	114.75	0.25963	3.7883	114.68	0.25687
120	6.2081	116.68	0.26964	5.1639	116.61	0.26591	4.4177	116.54	0.26274	3.8582	116.47	0.25998
130	6.3184	118.48	0.27272	5.2562	118.41	0.26899	4.4971	118.34	0.26582	3.9280	118.28	0.26307
140	6.4287	120.30	0.27577	5.3485	120.23	0.27204	4.5765	120.16	0.26888	3.9978	120.10	0.26613
150	6.5389	122.13	0.27880	5.4406	122.06	0.27507	4.6558	122.00	0.27191	4.0675	121.93	0.26916
160	6.6489	123.97	0.28180	5.5327	123.91	0.27808	4.7351	123.84	0.27492	4.1370	123.78	0.27218
170	6.7589	125.83	0.28478	5.6248	125.77	0.28106	4.8143	125.71	0.27791	4.2065	125.65	0.27516
180	6.8689	127.71	0.28774	5.7168	127.65	0.28402	4.8933	127.59	0.28087	4.2760	127.53	0.27813
190	6.9788	129.60	0.29067	5.8087	129.54	0.28696	4.9723	129.48	0.28381	4.3454	129.43	0.28107
200	7.0887	131.51	0.29359	5.9005	131.45	0.28987	5.0513	131.40	0.28672	4.4147	131.34	0.28399
210	7.1985	133.43	0.29648	5.9922	133.38	0.29277	5.1302	133.32	0.28962	4.4840	133.27	0.28689
220	7.3082	135.37	0.29935	6.0839	135.32	0.29564	5.2091	135.26	0.29250	4.5532	135.21	0.28977
230	7.4179	137.33	0.30221	6.1756	137.27	0.29850	5.2879	137.22	0.29535	4.6224	137.17	0.29263
240	7.5276	139.30	0.30504	6.2673	139.24	0.30134	5.3667	139.19	0.29819	4.6915	139.14	0.29546
250	—	—	—	6.3590	141.23	0.30416	5.4454	141.18	0.30102	4.7606	141.13	0.29829
260	—	—	—	—	—	—	5.5241	143.18	0.30382	4.8296	143.13	0.30109

Temp F	Pressure 18 psia Gage pressure 3.3 psig (Sat temp −19.6 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp −15.1 F)			Pressure 22 psia Gage pressure 7.3 psig (Sat temp −10.9 F)			Pressure 24 psia Gage pressure 9.3 psig (Sat temp −7.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
−10	2.6026	94.27	0.21442	2.3325	94.17	0.21216	2.1111	94.07	0.21010	—	—	—
0	2.6667	95.89	0.21797	2.3905	95.79	0.21572	2.1644	95.68	0.21367	1.9759	95.58	0.21179
10	2.7304	97.51	0.22147	2.4483	97.42	0.21923	2.2173	97.32	0.21719	2.0247	97.22	0.21531
20	2.7940	99.16	0.22494	2.5059	99.06	0.22270	2.2701	98.97	0.22066	2.0733	98.88	0.21879
30	2.8575	100.81	0.22836	2.5634	100.72	0.22612	2.3226	100.63	0.22409	2.1218	100.54	0.22223
40	2.9208	102.49	0.23174	2.6208	102.40	0.22951	2.3750	102.31	0.22748	2.1703	102.22	0.22562
50	2.9840	104.17	0.23508	2.6780	104.09	0.23286	2.4273	104.00	0.23084	2.2186	103.91	0.22898
60	3.0470	105.88	0.23838	2.7350	105.79	0.23617	2.4794	105.71	0.23415	2.2666	105.63	0.23230
70	3.1099	107.59	0.24166	2.7919	107.51	0.23945	2.5314	107.43	0.23743	2.3145	107.35	0.23559
80	3.1727	109.32	0.24490	2.8486	109.24	0.24269	2.5833	109.16	0.24068	2.3623	109.09	0.23884
90	3.2354	111.07	0.24810	2.9053	110.99	0.24590	2.6351	110.92	0.24389	2.4100	110.84	0.24206
100	3.2979	112.83	0.25127	2.9619	112.76	0.24908	2.6868	112.68	0.24708	2.4576	112.61	0.24524
110	3.3603	114.61	0.25442	3.0184	114.54	0.25223	2.7385	114.46	0.25023	2.5051	114.39	0.24840
120	3.4227	116.40	0.25754	3.0749	116.33	0.25535	2.7900	116.26	0.25335	2.5526	116.19	0.25153
130	3.4850	118.21	0.26063	3.1312	118.14	0.25844	2.8414	118.07	0.25645	2.5999	118.00	0.25463
140	3.5473	120.03	0.26369	3.1874	119.96	0.26151	2.8927	119.89	0.25952	2.6471	119.83	0.25770
150	3.6095	121.86	0.26673	3.2436	121.80	0.26455	2.9440	121.73	0.26256	2.6943	121.67	0.26074
160	3.6715	123.72	0.26974	3.2997	123.65	0.26756	2.9951	123.59	0.26558	2.7414	123.52	0.26377
170	3.7335	125.58	0.27273	3.3557	125.52	0.27055	3.0462	125.46	0.26858	2.7884	125.40	0.26676
180	3.7955	127.47	0.27570	3.4116	127.41	0.27352	3.0973	127.34	0.27155	2.8354	127.28	0.26974
190	3.8574	129.37	0.27864	3.4675	129.31	0.27647	3.1483	129.25	0.27450	2.8822	129.19	0.27269
200	3.9192	131.28	0.28156	3.5233	131.22	0.27939	3.1992	131.16	0.27742	2.9290	131.10	0.27561
210	3.9809	133.21	0.28446	3.5791	133.15	0.28230	3.2500	133.09	0.28033	2.9758	133.04	0.27852
220	4.0426	135.15	0.28735	3.6348	135.09	0.28518	3.3008	135.04	0.28321	3.0225	134.98	0.28141
230	4.1043	137.11	0.29021	3.6905	137.06	0.28804	3.3516	137.00	0.28607	3.0692	136.95	0.28427
240	4.1659	139.09	0.29305	3.7461	139.03	0.29089	3.4023	138.98	0.28892	3.1158	138.93	0.28712
250	4.2275	141.07	0.29587	3.8017	141.02	0.29371	3.4529	140.97	0.29175	3.1624	140.92	0.28995
260	4.2891	143.08	0.29868	3.8572	143.03	0.29652	3.5035	142.98	0.29456	3.2089	142.93	0.29276
270	4.3507	145.10	0.30147	3.9127	145.05	0.29931	3.5542	145.00	0.29735	3.2554	144.95	0.29556
280	—	—	—	—	—	—	—	—		3.3018	146.99	0.29833

Table 11. Carrene 7—Properties of Superheated Vapor (Continued)

emp F	Pressure 26 psia Gage pressure 11.3 psig (Sat temp −3.4 F)			Pressure 28 psia Gage pressure 13.3 psig (Sat temp 0.1 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 3.3 F)			Pressure 32 psia Gage pressure 17.3 psig (Sat temp 6.4 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
0	1.8162	95.48	0.21003	—	—	—	—	—	—	—	—	—
10	1.8617	97.12	0.21357	1.7219	97.02	0.21194	1.6008	96.92	0.21042	1.4947	96.82	0.20899
20	1.9069	98.78	0.21706	1.7643	98.68	0.21544	1.6406	98.58	0.21393	1.5323	98.49	0.21250
30	1.9520	100.45	0.22050	1.8064	100.35	0.21889	1.6801	100.26	0.21738	1.5696	100.16	0.21596
40	1.9970	102.13	0.22390	1.8484	102.04	0.22230	1.7196	101.95	0.22080	1.6068	101.86	0.21938
50	2.0418	103.83	0.22727	1.8902	103.74	0.22567	1.7589	103.65	0.22417	1.6438	103.56	0.22276
60	2.0865	105.54	0.23059	1.9318	105.45	0.22900	1.7980	105.37	0.22751	1.6807	105.28	0.22610
70	2.1310	107.26	0.23388	1.9733	107.18	0.23229	1.8369	107.10	0.23080	1.7175	107.02	0.22941
80	2.1752	109.01	0.23714	2.0147	108.92	0.23555	1.8758	108.84	0.23407	1.7541	108.76	0.23268
90	2.2194	110.76	0.24036	2.0560	110.68	0.23877	1.9146	110.61	0.23730	1.7906	110.53	0.23591
00	2.2636	112.53	0.24355	2.0972	112.46	0.24197	1.9532	112.38	0.24050	1.8270	112.30	0.23911
10	2.3077	114.32	0.24671	2.1384	114.24	0.24513	1.9917	114.17	0.24366	1.8633	114.09	0.24228
20	2.3516	116.11	0.24984	2.1794	116.04	0.24827	2.0301	115.97	0.24680	1.8995	115.90	0.24542
30	2.3955	117.93	0.25294	2.2203	117.86	0.25138	2.0685	117.79	0.24991	1.9356	117.72	0.24853
40	2.4393	119.76	0.25602	2.2611	119.69	0.25445	2.1068	119.62	0.25299	1.9716	119.55	0.25162
50	2.4830	121.60	0.25906	2.3019	121.53	0.25750	2.1449	121.47	0.25605	2.0076	121.40	0.25468
60	2.5266	123.46	0.26209	2.3426	123.39	0.26053	2.1830	123.33	0.25908	2.0435	123.26	0.25771
70	2.5702	125.34	0.26509	2.3831	125.27	0.26353	2.2211	125.21	0.26208	2.0792	125.14	0.26072
80	2.6137	127.23	0.26806	2.4236	127.16	0.26651	2.2591	127.10	0.26506	2.1149	127.04	0.26370
90	2.6572	129.13	0.27102	2.4641	129.06	0.26947	2.2969	129.00	0.26802	2.1506	128.94	0.26666
00	2.7005	131.04	0.27395	2.5045	130.99	0.27240	2.3347	130.93	0.27095	2.1862	130.87	0.26960
10	2.7438	132.98	0.27686	2.5449	132.92	0.27531	2.3725	132.86	0.27386	2.2217	132.81	0.27251
20	2.7870	134.93	0.27974	2.5852	134.87	0.27820	2.4103	134.81	0.27676	2.2572	134.76	0.27541
30	2.8302	136.89	0.28261	2.6254	136.84	0.28107	2.4480	136.78	0.27963	2.2926	136.73	0.27828
40	2.8734	138.87	0.28546	2.6656	138.82	0.28392	2.4856	138.76	0.28248	2.3280	138.71	0.28113
50	2.9165	140.86	0.28829	2.7057	140.81	0.28675	2.5232	140.76	0.28532	2.3633	140.71	0.28397
60	2.9595	142.88	0.29111	2.7458	142.82	0.28957	2.5607	142.77	0.28813	2.3986	142.72	0.28679
70	3.0025	144.90	0.29390	2.7859	144.85	0.29236	2.5981	144.80	0.29093	2.4338	144.75	0.28959
80	3.0455	146.95	0.29668	2.8259	146.89	0.29514	2.6355	146.84	0.29371	2.4690	146.80	0.29237
90	—	—	—	2.8659	148.95	0.29790	2.6730	148.90	0.29648	2.5041	148.85	0.29513

emp F	Pressure 34 psia Gage pressure 19.3 psig (Sat temp 9.4 F)			Pressure 36 psia Gage pressure 21.3 psig (Sat temp 12.2 F)			Pressure 38 psia Gage pressure 23.3 psig (Sat temp 14.9 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 17.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
10	1.4010	96.72	0.20763	—	—	—	—	—	—	—	—	—
20	1.4366	98.39	0.21115	1.3516	98.29	0.20986	1.2756	98.19	0.20864	1.2070	98.09	0.20747
30	1.4720	100.07	0.21462	1.3853	99.97	0.21334	1.3078	99.88	0.21213	1.2378	99.78	0.21097
40	1.5073	101.77	0.21804	1.4188	101.67	0.21677	1.3397	101.58	0.21557	1.2683	101.49	0.21441
50	1.5424	103.47	0.22143	1.4521	103.38	0.22017	1.3714	103.29	0.21896	1.2987	103.20	0.21782
60	1.5773	105.20	0.22477	1.4853	105.11	0.22352	1.4030	105.02	0.22232	1.3290	104.93	0.22118
70	1.6121	106.93	0.22809	1.5183	106.85	0.22683	1.4345	106.76	0.22564	1.3591	106.68	0.22451
80	1.6468	108.68	0.23136	1.5512	108.60	0.23011	1.4659	108.52	0.22892	1.3889	108.43	0.22779
90	1.6813	110.45	0.23460	1.5840	110.37	0.23335	1.4971	110.29	0.23217	1.4187	110.21	0.23104
00	1.7157	112.22	0.23780	1.6167	112.15	0.23656	1.5282	112.07	0.23539	1.4485	112.00	0.23426
10	1.7500	114.02	0.24098	1.6493	113.94	0.23974	1.5592	113.87	0.23857	1.4781	113.79	0.23745
20	1.7842	115.82	0.24413	1.6817	115.75	0.24289	1.5901	115.68	0.24172	1.5075	115.60	0.24060
30	1.8183	117.65	0.24724	1.7141	117.58	0.24601	1.6209	117.51	0.24484	1.5369	117.43	0.24373
40	1.8523	119.49	0.25032	1.7464	119.41	0.24910	1.6516	119.34	0.24794	1.5662	119.28	0.24683
50	1.8863	121.34	0.25338	1.7786	121.27	0.25216	1.6822	121.20	0.25100	1.5954	121.13	0.24990
60	1.9202	123.20	0.25642	1.8107	123.14	0.25520	1.7127	123.07	0.25404	1.6246	123.00	0.25294
70	1.9540	125.08	0.25943	1.8427	125.02	0.25821	1.7432	124.95	0.25706	1.6537	124.89	0.25596
80	1.9877	126.98	0.26241	1.8747	126.91	0.26120	1.7736	126.85	0.26005	1.6826	126.79	0.25895
90	2.0214	128.88	0.26538	1.9066	128.82	0.26417	1.8039	128.76	0.26301	1.7115	128.71	0.26192
00	2.0550	130.81	0.26832	1.9385	130.75	0.26711	1.8342	130.69	0.26596	1.7404	130.63	0.26487
10	2.0886	132.75	0.27123	1.9703	132.69	0.27003	1.8644	132.63	0.26888	1.7692	132.57	0.26779
20	2.1221	134.70	0.27413	2.0020	134.65	0.27292	1.8946	134.59	0.27178	1.7979	134.53	0.27069
30	2.1555	136.67	0.27701	2.0337	136.62	0.27580	1.9247	136.56	0.27466	1.8265	136.51	0.27357
40	2.1889	138.66	0.27986	2.0653	138.61	0.27866	1.9547	138.55	0.27752	1.8551	138.50	0.27644
50	2.2222	140.65	0.28270	2.0969	140.60	0.28150	1.9847	140.55	0.28036	1.8837	140.50	0.27928
60	2.2555	142.67	0.28552	2.1284	142.62	0.28432	2.0146	142.57	0.28318	1.9123	142.52	0.28210
70	2.2887	144.70	0.28832	2.1599	144.65	0.28712	2.0445	144.60	0.28599	1.9408	144.55	0.28491
80	2.3219	146.75	0.29110	2.1913	146.70	0.28991	2.0744	146.65	0.28877	1.9692	146.60	0.28770
90	2.3551	148.80	0.29387	2.2227	148.76	0.29268	2.1042	148.71	0.29154	1.9976	148.66	0.29047
00	—	—	—	2.2541	150.83	0.29543	2.1340	150.79	0.29429	2.0260	150.74	0.29322

Table 11. Carrene 7—Properties of Superheated Vapor (Continued)

Temp F	Pressure 42 psia Gage pressure 27.3 psig (Sat temp 20.0 F)			Pressure 44 psia Gage pressure 29.3 psig (Sat temp 22.4 F)			Pressure 46 psia Gage pressure 31.3 psig (Sat temp 24.7 F)			Pressure 48 psia Gage pressure 33.3 psig (Sat temp 26.9 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
20	1.1450	97.99	0.20636	—	—	—	—	—	—	—	—	—
30	1.1744	99.68	0.20985	1.1169	99.58	0.20879	1.0644	99.49	0.20776	1.0162	99.38	0.20677
40	1.2036	101.39	0.21330	1.1451	101.30	0.21225	1.0915	101.20	0.21123	1.0424	101.10	0.21025
50	1.2328	103.11	0.21671	1.1731	103.02	0.21566	1.1184	102.93	0.21465	1.0683	102.83	0.21368
60	1.2619	104.85	0.22009	1.2009	104.76	0.21904	1.1451	104.67	0.21803	1.0941	104.58	0.21706
70	1.2908	106.59	0.22342	1.2286	106.51	0.22238	1.1718	106.42	0.22137	1.1198	106.33	0.22041
80	1.3194	108.36	0.22671	1.2561	108.27	0.22567	1.1983	108.19	0.22468	1.1453	108.10	0.22372
90	1.3479	110.13	0.22997	1.2835	110.05	0.22893	1.2246	109.97	0.22794	1.1707	109.89	0.22699
100	1.3762	111.92	0.23319	1.3108	111.84	0.23216	1.2508	111.76	0.23117	1.1959	111.68	0.23022
110	1.4045	113.72	0.23638	1.3380	113.64	0.23536	1.2769	113.56	0.23437	1.2210	113.49	0.23342
120	1.4328	115.53	0.23954	1.3650	115.46	0.23852	1.3029	115.38	0.23754	1.2460	115.31	0.23660
130	1.4609	117.36	0.24267	1.3919	117.29	0.24165	1.3288	117.22	0.24067	1.2710	117.15	0.23974
140	1.4890	119.21	0.24577	1.4187	119.14	0.24475	1.3546	119.07	0.24378	1.2959	119.00	0.24285
150	1.5169	121.06	0.24884	1.4455	120.99	0.24783	1.3803	120.93	0.24686	1.3206	120.86	0.24593
160	1.5447	122.93	0.25188	1.4722	122.87	0.25088	1.4059	122.80	0.24991	1.3452	122.74	0.24898
170	1.5725	124.82	0.25491	1.4988	124.76	0.25390	1.4315	124.69	0.25294	1.3698	124.63	0.25201
180	1.6002	126.72	0.25790	1.5253	126.66	0.25690	1.4570	126.60	0.25594	1.3943	126.53	0.25502
190	1.6278	128.64	0.26087	1.5518	128.58	0.25988	1.4824	128.52	0.25892	1.4188	128.46	0.25800
200	1.6554	130.57	0.26382	1.5782	130.51	0.26282	1.5077	130.45	0.26187	1.4432	130.39	0.26095
210	1.6829	132.52	0.26675	1.6046	132.46	0.26575	1.5330	132.40	0.26480	1.4674	132.34	0.26388
220	1.7103	134.48	0.26965	1.6308	134.42	0.26866	1.5582	134.36	0.26771	1.4916	134.31	0.26679
230	1.7377	136.45	0.27254	1.6570	136.40	0.27155	1.5834	136.35	0.27060	1.5158	136.28	0.26968
240	1.7651	138.44	0.27540	1.6832	138.39	0.27441	1.6085	138.33	0.27346	1.5400	138.28	0.27256
250	1.7924	140.44	0.27825	1.7094	140.39	0.27726	1.6336	140.34	0.27631	1.5641	140.29	0.27541
260	1.8197	142.46	0.28107	1.7355	142.41	0.28009	1.6586	142.36	0.27914	1.5881	142.31	0.27823
270	1.8468	144.50	0.28388	1.7615	144.45	0.28290	1.6836	144.40	0.28195	1.6121	144.35	0.28105
280	1.8739	146.55	0.28667	1.7875	146.50	0.28569	1.7085	146.45	0.28475	1.6360	146.40	0.28384
290	1.9010	148.61	0.28944	1.8134	148.56	0.28846	1.7333	148.51	0.28752	1.6599	148.47	0.28662
300	1.9282	150.69	0.29219	1.8393	150.65	0.29121	1.7582	150.60	0.29028	1.6838	150.55	0.28938
310	—	—	—	1.8652	152.74	0.29396	1.7830	152.69	0.29302	1.7076	152.65	0.29212

Temp F	Pressure 50 psia Gage pressure 35.3 psig (Sat temp 29.1 F)			Pressure 52 psia Gage pressure 37.3 psig (Sat temp 31.2 F)			Pressure 54 psia Gage pressure 39.3 psig (Sat temp 33.3 F)			Pressure 56 psia Gage pressure 41.3 psig (Sat temp 35.3 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
30	0.9718	99.28	0.20582	—	—	—	—	—	—	—	—	—
40	0.9971	101.01	0.20930	0.9554	100.91	0.20838	0.9166	100.81	0.20749	0.8807	100.71	0.20663
50	1.0222	102.74	0.21273	0.9796	102.65	0.21182	0.9402	102.55	0.21094	0.9035	102.46	0.21008
60	1.0471	104.49	0.21612	1.0037	104.40	0.21522	0.9635	104.30	0.21435	0.9262	104.21	0.21350
70	1.0719	106.25	0.21948	1.0277	106.16	0.21858	0.9867	106.07	0.21771	0.9487	105.97	0.21687
80	1.0966	108.02	0.22279	1.0515	107.93	0.22190	1.0098	107.85	0.22103	0.9711	107.76	0.22020
90	1.1211	109.80	0.22607	1.0752	109.72	0.22518	1.0328	109.64	0.22432	0.9933	109.56	0.22349
100	1.1454	111.60	0.22931	1.0988	111.52	0.22842	1.0556	111.44	0.22757	1.0154	111.36	0.22674
110	1.1696	113.41	0.23251	1.1222	113.33	0.23163	1.0783	113.26	0.23078	1.0374	113.18	0.22996
120	1.1973	115.24	0.23569	1.1454	115.16	0.23481	1.1007	115.08	0.23396	1.0592	115.01	0.23315
130	1.2178	117.07	0.23883	1.1686	117.00	0.23796	1.1231	116.93	0.23712	1.0810	116.85	0.23630
140	1.2418	118.93	0.24195	1.1918	118.85	0.24108	1.1455	118.79	0.24024	1.1026	118.71	0.23943
150	1.2656	120.79	0.24503	1.2149	120.72	0.24417	1.1678	120.66	0.24333	1.1241	120.58	0.24252
160	1.2893	122.67	0.24809	1.2378	122.60	0.24723	1.1900	122.54	0.24639	1.1456	122.47	0.24559
170	1.3130	124.56	0.25112	1.2607	124.50	0.25026	1.2121	124.44	0.24943	1.1670	124.37	0.24863
180	1.3366	126.47	0.25413	1.2834	126.41	0.25327	1.2341	126.34	0.25244	1.1883	126.28	0.25164
190	1.3601	128.39	0.25711	1.3061	128.33	0.25625	1.2560	128.27	0.25543	1.2095	128.21	0.25463
200	1.3836	130.33	0.26007	1.3287	130.27	0.25921	1.2779	130.21	0.25839	1.2307	130.15	0.25760
210	1.4070	132.28	0.26300	1.3513	132.22	0.26215	1.2997	132.16	0.26133	1.2518	132.11	0.26054
220	1.4304	134.25	0.26592	1.3738	134.19	0.26507	1.3215	134.13	0.26425	1.2728	134.07	0.26346
230	1.4537	136.23	0.26881	1.3963	136.18	0.26796	1.3432	136.12	0.26715	1.2938	136.06	0.26636
240	1.4769	138.22	0.27168	1.4187	138.17	0.27083	1.3648	138.11	0.27002	1.3148	138.06	0.26923
250	1.5000	140.23	0.27453	1.4410	140.18	0.27369	1.3864	140.12	0.27288	1.3357	140.07	0.27209
260	1.5232	142.26	0.27736	1.4633	142.20	0.27652	1.4079	142.15	0.27571	1.3565	142.10	0.27493
270	1.5463	144.30	0.28018	1.4856	144.24	0.27934	1.4294	144.19	0.27853	1.3773	144.15	0.27775
280	1.5694	146.35	0.28297	1.5079	146.30	0.28214	1.4508	146.25	0.28133	1.3980	146.20	0.28055
290	1.5924	148.42	0.28575	1.5301	148.37	0.28491	1.4723	148.32	0.28411	1.4187	148.27	0.28333
300	1.6153	150.50	0.28851	1.5522	150.45	0.28768	1.4937	150.41	0.28687	1.4393	150.36	0.28609
310	1.6382	152.60	0.29125	1.5743	152.55	0.29042	1.5150	152.51	0.28962	1.4599	152.46	0.28884
320	—	—	—	1.5963	154.67	0.29315	1.5363	154.62	0.29235	1.4805	154.58	0.29158

Table 11. Carrene 7—Properties of Superheated Vapor (Continued)

Temp F	Pressure 58 psia Gage pressure 43.3 psig (Sat temp 37.2 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 39.1 F)			Pressure 62 psia Gage pressure 47.3 psig (Sat temp 41.0 F)			Pressure 64 psia Gage pressure 49.3 psig (Sat temp 42.8 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
40	0.8471	100.61	0.20579	0.8159	100.52	0.20497	—	—	—	—	—	—
50	0.8693	102.36	0.20925	0.8375	102.26	0.20845	0.8075	102.17	0.20765	0.7796	102.07	0.20689
60	0.8913	104.12	0.21267	0.8589	104.03	0.21187	0.8284	103.93	0.21109	0.8000	103.84	0.21033
70	0.9132	105.89	0.21605	0.8802	105.80	0.21525	0.8492	105.71	0.21448	0.8202	105.62	0.21372
80	0.9350	107.67	0.21938	0.9013	107.59	0.21859	0.8698	107.50	0.21783	0.8403	107.41	0.21708
90	0.9566	109.47	0.22268	0.9223	109.39	0.22189	0.8902	109.30	0.22113	0.8602	109.22	0.22039
100	0.9780	111.28	0.22594	0.9431	111.20	0.22516	0.9104	111.12	0.22440	0.8799	111.04	0.22366
110	0.9993	113.10	0.22916	0.9638	113.02	0.22839	0.9305	112.94	0.22763	0.8994	112.86	0.22690
120	1.0205	114.93	0.23235	0.9843	114.86	0.23158	0.9505	114.78	0.23083	0.9189	114.71	0.23010
130	1.0416	116.78	0.23551	1.0048	116.71	0.23474	0.9705	116.64	0.23400	0.9383	116.56	0.23327
140	1.0625	118.64	0.23864	1.0253	118.57	0.23787	0.9904	118.50	0.23713	0.9576	118.43	0.23641
150	1.0834	120.51	0.24173	1.0456	120.44	0.24098	1.0101	120.37	0.24024	0.9768	120.30	0.23952
160	1.1043	122.40	0.24481	1.0657	122.33	0.24405	1.0296	122.27	0.24331	0.9959	122.20	0.24260
170	1.1251	124.30	0.24785	1.0858	124.24	0.24710	1.0491	124.17	0.24636	1.0149	124.10	0.24565
180	1.1457	126.22	0.25087	1.1058	126.15	0.25012	1.0686	126.09	0.24939	1.0338	126.02	0.24868
190	1.1662	128.15	0.25386	1.1258	128.08	0.25311	1.0880	128.02	0.25238	1.0526	127.96	0.25168
200	1.1867	130.09	0.25682	1.1457	130.03	0.25608	1.1073	129.97	0.25535	1.0714	129.91	0.25465
210	1.2071	132.05	0.25977	1.1655	131.99	0.25903	1.1266	131.93	0.25830	1.0901	131.87	0.25760
220	1.2275	134.02	0.26269	1.1853	133.96	0.26195	1.1457	133.90	0.26123	1.1087	133.84	0.26053
230	1.2478	136.00	0.26559	1.2050	135.95	0.26485	1.1648	135.89	0.26413	1.1273	135.84	0.26344
240	1.2681	138.00	0.26847	1.2246	137.95	0.26773	1.1839	137.89	0.26702	1.1458	137.84	0.26632
250	1.2883	140.02	0.27133	1.2442	139.96	0.27059	1.2030	139.91	0.26988	1.1643	139.86	0.26919
260	1.3084	142.05	0.27417	1.2637	142.00	0.27343	1.2219	141.95	0.27272	1.1828	141.89	0.27203
270	1.3285	144.09	0.27699	1.2833	144.04	0.27626	1.2408	143.99	0.27555	1.2012	143.94	0.27486
280	1.3487	146.15	0.27979	1.3028	146.10	0.27906	1.2597	146.05	0.27835	1.2196	146.00	0.27767
290	1.3688	148.22	0.28258	1.3222	148.17	0.28185	1.2785	148.13	0.28114	1.2377	148.08	0.28045
300	1.3887	150.31	0.28534	1.3415	150.26	0.28462	1.2973	150.22	0.28391	1.2560	150.17	0.28322
310	1.4086	152.42	0.28809	1.3608	152.37	0.28737	1.3161	152.33	0.28666	1.2742	152.28	0.28598
320	1.4285	154.53	0.29083	1.3801	154.49	0.29010	1.3348	154.45	0.28940	1.2924	154.40	0.28872
330	—	—	—	—	—	—	1.3534	156.58	0.29212	1.3105	156.53	0.29144

Temp F	Pressure 66 psia Gage pressure 51.3 psig (Sat temp 44.5 F)			Pressure 68 psia Gage pressure 53.3 psig (Sat temp 46.3 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 47.9 F)			Pressure 80 psia Gage pressure 65.3 psig (Sat temp 55.8 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
50	0.7533	101.97	0.20614	0.7284	101.87	0.20540	0.7051	101.77	0.20469	—	—	—
60	0.7732	103.75	0.20959	0.7478	103.65	0.20886	0.7242	103.56	0.20816	0.6234	103.07	0.20485
70	0.7929	105.53	0.21299	0.7671	105.44	0.21227	0.7430	105.35	0.21157	0.6400	104.88	0.20829
80	0.8124	107.33	0.21635	0.7862	107.24	0.21563	0.7616	107.15	0.21494	0.6566	106.70	0.21168
90	0.8318	109.13	0.21966	0.8052	109.05	0.21896	0.7801	108.96	0.21827	0.6732	108.53	0.21504
100	0.8511	110.95	0.22294	0.8240	110.87	0.22224	0.7984	110.79	0.22156	0.6897	110.37	0.21836
110	0.8702	112.79	0.22619	0.8426	112.70	0.22549	0.8166	112.62	0.22481	0.7061	112.22	0.22163
120	0.8892	114.63	0.22939	0.8611	114.55	0.22870	0.8347	114.47	0.22803	0.7223	114.09	0.22487
130	0.9080	116.48	0.23257	0.8795	116.41	0.23188	0.8526	116.33	0.23121	0.7384	115.96	0.22808
140	0.9268	118.35	0.23571	0.8978	118.28	0.23503	0.8704	118.21	0.23436	0.7543	117.84	0.23125
150	0.9455	120.23	0.23882	0.9160	120.17	0.23814	0.8882	120.09	0.23748	0.7701	119.74	0.23438
160	0.9640	122.13	0.24190	0.9341	122.06	0.24123	0.9059	121.99	0.24057	0.7859	121.65	0.23749
170	0.9825	124.04	0.24496	0.9522	123.97	0.24429	0.9235	123.90	0.24363	0.8016	123.57	0.24057
180	1.0009	125.96	0.24799	0.9702	125.90	0.24732	0.9410	125.83	0.24666	0.8172	125.50	0.24362
190	1.0193	127.90	0.25099	0.9880	127.83	0.25033	0.9584	127.77	0.24967	0.8327	127.45	0.24664
200	1.0375	129.85	0.25397	1.0058	129.79	0.25330	0.9757	129.72	0.25265	0.8482	129.42	0.24964
210	1.0057	131.81	0.25692	1.0235	131.75	0.25626	0.9929	131.69	0.25561	0.8638	131.39	0.25261
220	1.0739	133.79	0.25985	1.0411	133.73	0.25919	1.0101	133.67	0.25854	0.8790	133.38	0.25556
230	1.0920	135.78	0.26276	1.0586	135.72	0.26210	1.0273	135.67	0.26146	0.8941	135.38	0.25848
240	1.1100	137.79	0.26565	1.0762	137.73	0.26499	1.0444	137.67	0.26435	0.9092	137.40	0.26138
250	1.1279	139.80	0.26851	1.0937	139.75	0.26786	1.0614	139.70	0.26722	0.9243	139.42	0.26426
260	1.1458	141.84	0.27136	1.1111	141.79	0.27071	1.0784	141.73	0.27007	0.9394	141.47	0.26712
270	1.1637	143.89	0.27419	1.1285	143.84	0.27354	1.0954	143.78	0.27290	0.9544	143.53	0.26996
280	1.1815	145.96	0.27700	1.1459	145.90	0.27635	1.1123	145.85	0.27571	0.9694	145.60	0.27278
290	1.1993	148.03	0.27979	1.1632	147.98	0.27914	1.1292	147.93	0.27851	0.9843	147.69	0.27558
300	1.2171	150.12	0.28256	1.1805	150.07	0.28191	1.1459	150.03	0.28128	0.9992	149.79	0.27836
310	1.2348	152.23	0.28532	1.1977	152.18	0.28467	1.1626	152.14	0.28404	1.0140	151.90	0.28113
320	1.2524	154.35	0.28805	1.2149	154.31	0.28741	1.1793	154.26	0.28678	1.0288	154.03	0.28388
330	1.2700	156.49	0.29078	1.2320	156.44	0.29013	1.1967	156.40	0.28952	1.0436	156.18	0.28661
340	—	—	—	—	—	—	—	—	—	1.0583	158.33	0.28932

Table 11. Carrene 7—Properties of Superheated Vapor (Continued)

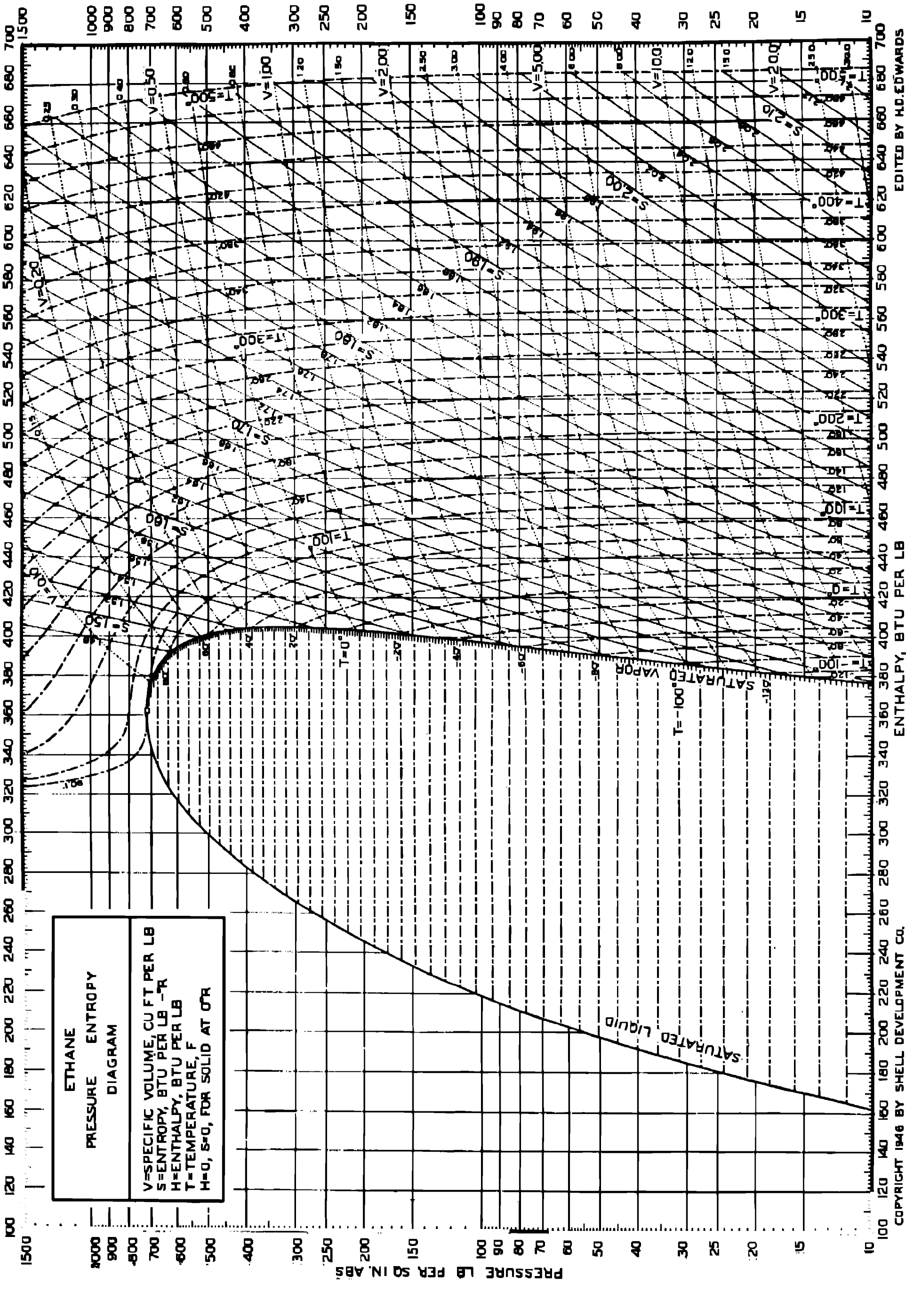
Temp F	Pressure 90 psia Gage pressure 75.3 psig (Sat temp 63.0 F)			Pressure 100 psia Gage pressure 85.3 psig (Sat temp 69.7 F)			Pressure 110 psia Gage pressure 95.3 psig (Sat temp 75.8 F)			Pressure 120 psia Gage pressure 105.3 psig (Sat temp 81.6 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
60	0.5434	102.56	0.20178	—	—	—	—	—	—	—	—	—
70	0.5591	104.39	0.20527	0.4944	103.89	0.20249	—	—	—	—	—	—
80	0.5745	106.23	0.20871	0.5087	105.75	0.20597	0.4539	105.25	0.20337	—	—	—
90	0.5898	108.08	0.21210	0.5229	107.62	0.20940	0.4677	107.15	0.20686	0.4220	106.66	0.20449
100	0.6050	109.94	0.21545	0.5369	109.50	0.21278	0.4811	109.04	0.21029	0.4345	108.58	0.20795
110	0.6200	111.81	0.21876	0.5508	111.38	0.21612	0.4941	110.95	0.21366	0.4468	110.50	0.21135
120	0.6347	113.68	0.22202	0.5645	113.27	0.21941	0.5069	112.86	0.21698	0.4589	112.43	0.21471
130	0.6493	115.57	0.22525	0.5780	115.17	0.22266	0.5195	114.77	0.22026	0.4707	114.36	0.21801
140	0.6639	117.47	0.22844	0.5914	117.09	0.22587	0.5319	116.70	0.22350	0.4824	116.30	0.22128
150	0.6783	119.37	0.23160	0.6046	119.01	0.22905	0.5442	118.63	0.22669	0.4939	118.25	0.22450
160	0.6925	121.29	0.23472	0.6177	120.94	0.23219	0.5565	120.58	0.22986	0.5053	120.21	0.22768
170	0.7067	123.23	0.23781	0.6308	122.88	0.23531	0.5686	122.53	0.23299	0.5167	122.18	0.23084
180	0.7209	125.17	0.24088	0.6437	124.84	0.23838	0.5805	124.50	0.23609	0.5279	124.16	0.23395
190	0.7349	127.13	0.24392	0.6566	126.81	0.24144	0.5924	126.48	0.23915	0.5390	126.15	0.23703
200	0.7489	129.10	0.24693	0.6694	128.79	0.24446	0.6043	128.47	0.24219	0.5500	128.15	0.24009
210	0.7628	131.09	0.24991	0.6821	130.78	0.24746	0.6161	130.47	0.24521	0.5610	130.16	0.24311
220	0.7766	133.08	0.25287	0.6947	132.78	0.25043	0.6277	132.49	0.24819	0.5719	132.18	0.24611
230	0.7903	135.09	0.25581	0.7072	134.80	0.25338	0.6393	134.51	0.25115	0.5827	134.21	0.24909
240	0.8040	137.12	0.25872	0.7198	136.84	0.25630	0.6509	136.55	0.25409	0.5934	136.26	0.25203
250	0.8176	139.15	0.26161	0.7323	138.88	0.25921	0.6623	138.60	0.25699	0.6041	138.32	0.25496
260	0.8312	141.20	0.26448	0.7447	140.94	0.26208	0.6737	140.66	0.25988	0.6148	140.39	0.25786
270	0.8447	143.27	0.26733	0.7570	143.01	0.26494	0.6851	142.74	0.26275	0.6254	142.48	0.26074
280	0.8582	145.35	0.27015	0.7693	145.09	0.26778	0.6965	144.84	0.26560	0.6359	144.58	0.26359
290	0.8716	147.44	0.27296	0.7815	147.19	0.27060	0.7077	146.94	0.26843	0.6463	146.69	0.26642
300	0.8850	149.54	0.27575	0.7937	149.30	0.27339	0.7189	149.06	0.27123	0.6567	148.81	0.26924
310	0.8984	151.66	0.27853	0.8058	151.43	0.27617	0.7301	151.19	0.27402	0.6671	150.95	0.27204
320	0.9118	153.80	0.28129	0.8179	153.57	0.27894	0.7413	153.34	0.27679	0.6774	153.10	0.27481
330	0.9251	155.95	0.28403	0.8300	155.72	0.28168	0.7524	155.50	0.27955	0.6877	155.27	0.27757
340	0.9383	158.11	0.28675	0.8421	157.89	0.28441	0.7634	157.67	0.28228	0.6979	157.45	0.28032
350	—	—	—	0.8541	160.08	0.28713	0.7744	159.86	0.28500	0.7081	159.64	0.28304
360	—	—	—	—	—	—	0.7854	162.06	0.28770	0.7183	161.85	0.28575
370	—	—	—	—	—	—	—	—	—	0.7284	164.07	0.28844

Temp F	Pressure 130 psia Gage pressure 115.3 psig (Sat temp 87.0 F)			Pressure 140 psia Gage pressure 125.3 psig (Sat temp 92.1 F)			Pressure 150 psia Gage pressure 135.3 psig (Sat temp 96.9 F)			Pressure 160 psia Gage pressure 145.3 psig (Sat temp 101.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
90	0.3829	106.16	0.20222	—	—	—	—	—	—	—	—	—
100	0.3948	108.10	0.20572	0.3606	107.61	0.20359	0.3308	107.09	0.20153	—	—	—
110	0.4065	110.04	0.20917	0.3718	109.57	0.20707	0.3416	109.09	0.20506	0.3149	108.58	0.20311
120	0.4180	111.99	0.21256	0.3827	111.54	0.21050	0.3521	111.08	0.20853	0.3252	110.61	0.20663
130	0.4292	113.94	0.21589	0.3935	113.51	0.21387	0.3624	113.07	0.21193	0.3353	112.62	0.21008
140	0.4403	115.90	0.21918	0.4041	115.49	0.21719	0.3726	115.07	0.21529	0.3451	114.64	0.21347
150	0.4512	117.86	0.22243	0.4145	117.47	0.22047	0.3826	117.06	0.21860	0.3546	116.65	0.21680
160	0.4620	119.84	0.22564	0.4247	119.46	0.22370	0.3923	119.07	0.22185	0.3640	118.67	0.22009
170	0.4726	121.82	0.22880	0.4348	121.45	0.22689	0.4020	121.08	0.22507	0.3733	120.70	0.22333
180	0.4832	123.81	0.23194	0.4449	123.45	0.23005	0.4116	123.09	0.22825	0.3825	122.73	0.22653
190	0.4938	125.81	0.23505	0.4548	125.47	0.23317	0.4210	125.12	0.23139	0.3915	124.77	0.22970
200	0.5042	127.82	0.23812	0.4646	127.49	0.23626	0.4303	127.15	0.23449	0.4004	126.81	0.23282
210	0.5144	129.84	0.24116	0.4743	129.52	0.23931	0.4395	129.19	0.23756	0.4092	128.87	0.23591
220	0.5246	131.87	0.24417	0.4839	131.56	0.24234	0.4487	131.25	0.24061	0.4179	130.93	0.23896
230	0.5347	133.92	0.24715	0.4935	133.62	0.24534	0.4578	133.31	0.24362	0.4265	133.00	0.24199
240	0.5448	135.97	0.25011	0.5031	135.68	0.24831	0.4667	135.38	0.24660	0.4351	135.09	0.24499
250	0.5548	138.04	0.25305	0.5125	137.76	0.25126	0.4756	137.47	0.24956	0.4436	137.18	0.24796
260	0.5647	140.12	0.25596	0.5218	139.84	0.25418	0.4846	139.57	0.25250	0.4520	139.29	0.25090
270	0.5746	142.21	0.25885	0.5311	141.94	0.25708	0.4934	141.67	0.25541	0.4604	141.40	0.25383
280	0.5844	144.32	0.26171	0.5404	144.06	0.25996	0.5022	143.79	0.25830	0.4687	143.53	0.25672
290	0.5942	146.44	0.26456	0.5496	146.18	0.26281	0.5109	145.93	0.26116	0.4769	145.67	0.25959
300	0.6039	148.57	0.26738	0.5587	148.32	0.26564	0.5195	148.07	0.26400	0.4851	147.82	0.26244
310	0.6136	150.71	0.27019	0.5678	150.47	0.26845	0.5281	150.23	0.26682	0.4933	149.98	0.26528
320	0.6233	152.87	0.27297	0.5768	152.63	0.27124	0.5366	152.40	0.26962	0.5014	152.16	0.26808
330	0.6329	155.04	0.27574	0.5858	154.81	0.27402	0.5451	154.58	0.27240	0.5095	154.35	0.27087
340	0.6424	157.23	0.27849	0.5949	157.00	0.27678	0.5536	156.78	0.27517	0.5175	156.55	0.27365
350	0.6519	159.42	0.28122	0.6038	159.21	0.27952	0.5620	158.99	0.27791	0.5255	158.77	0.27640
360	0.6614	161.64	0.28393	0.6126	161.43	0.28223	0.5704	161.21	0.28064	0.5334	160.99	0.27913
370	0.6709	163.86	0.28663	0.6215	163.65	0.28494	0.5787	163.45	0.28335	0.5413	163.24	0.28185
380	—	—	—	0.6304	165.90	0.28763	0.5870	165.69	0.28604	0.5492	165.49	0.28455
390	—	—	—	—	—	—	—	—	—	0.5571	167.76	0.28724

Table 11. Carrene 7—Properties of Superheated Vapor (Concluded)

Temp F	Pressure 170 psia Gage pressure 155.3 psig (Sat temp 105.9 F)			Pressure 180 psia Gage pressure 165.3 psig (Sat temp 110.2 F)			Pressure 190 psia Gage pressure 175.3 psig (Sat temp 114.2 F)			Pressure 200 psia Gage pressure 185.3 psig (Sat temp 118.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
110	0.2913	108.07	0.20122	—	—	—	—	—	—	—	—	—
120	0.3012	110.11	0.20477	0.2800	109.61	0.20299	0.2606	109.08	0.20116	0.2432	108.54	0.19948
130	0.3110	112.16	0.20827	0.2892	111.67	0.20650	0.2698	111.18	0.20479	0.2524	110.68	0.20313
140	0.3206	114.20	0.21171	0.2983	113.74	0.20996	0.2789	113.27	0.20831	0.2612	112.80	0.20669
150	0.3299	116.23	0.21508	0.3074	115.79	0.21338	0.2878	115.36	0.21177	0.2698	114.90	0.21018
160	0.3390	118.27	0.21840	0.3165	117.86	0.21675	0.2964	117.44	0.21515	0.2782	117.01	0.21360
170	0.3479	120.31	0.22166	0.3252	119.92	0.22005	0.3048	119.52	0.21848	0.2864	119.11	0.21697
180	0.3567	122.36	0.22489	0.3336	121.98	0.22329	0.3130	121.60	0.22176	0.2944	121.21	0.22027
190	0.3654	124.41	0.22807	0.3419	124.05	0.22649	0.3210	123.68	0.22498	0.3023	123.31	0.22353
200	0.3740	126.47	0.23122	0.3502	126.12	0.22966	0.3290	125.77	0.22817	0.3100	125.41	0.22673
210	0.3824	128.54	0.23432	0.3583	128.20	0.23279	0.3369	127.86	0.23132	0.3176	127.51	0.22990
220	0.3907	130.61	0.23739	0.3664	130.29	0.23588	0.3446	129.96	0.23443	0.3251	129.62	0.23303
230	0.3989	132.69	0.24043	0.3744	132.38	0.23894	0.3523	132.06	0.23751	0.3325	131.74	0.23613
240	0.4071	134.79	0.24345	0.3822	134.48	0.24197	0.3599	134.18	0.24055	0.3397	133.87	0.23918
250	0.4152	136.89	0.24643	0.3899	136.59	0.24496	0.3674	136.30	0.24356	0.3469	136.00	0.24220
260	0.4233	139.00	0.24939	0.3976	138.72	0.24793	0.3748	138.43	0.24655	0.3541	138.14	0.24520
270	0.4314	141.13	0.25233	0.4053	140.85	0.25088	0.3822	140.57	0.24951	0.3612	140.29	0.24817
280	0.4393	143.26	0.25523	0.4129	142.99	0.25380	0.3895	142.73	0.25244	0.3682	142.45	0.25111
290	0.4471	145.41	0.25811	0.4204	145.15	0.25669	0.3966	144.69	0.25533	0.3752	144.62	0.25403
300	0.4549	147.57	0.26097	0.4278	147.31	0.25955	0.4037	147.06	0.25821	0.3821	146.80	0.25692
310	0.4627	149.74	0.26381	0.4352	149.49	0.26240	0.4108	149.24	0.26106	0.3889	149.00	0.25978
320	0.4704	151.92	0.26663	0.4426	151.68	0.26523	0.4179	151.44	0.26390	0.3956	151.20	0.26262
330	0.4780	154.12	0.26942	0.4500	153.88	0.26804	0.4249	153.65	0.26671	0.4023	153.41	0.26544
340	0.4856	156.32	0.27220	0.4573	156.10	0.27083	0.4319	155.87	0.26951	0.4091	155.64	0.26825
350	0.4933	158.55	0.27497	0.4645	158.32	0.27359	0.4389	158.10	0.27229	0.4158	157.88	0.27103
360	0.5008	160.78	0.27770	0.4717	160.56	0.27634	0.4458	160.35	0.27504	0.4224	160.13	0.27380
370	0.5083	163.03	0.28043	0.4789	162.82	0.27907	0.4526	162.60	0.27778	0.4290	162.39	0.27654
380	0.5158	165.29	0.28313	0.4860	165.08	0.28178	0.4593	164.87	0.28049	0.4355	164.67	0.27927
390	0.5233	167.56	0.28583	0.4931	167.36	0.28448	0.4661	167.16	0.28319	0.4420	166.95	0.28198
400	—	—	—	0.5002	169.65	0.28716	0.4729	169.45	0.28588	0.4484	169.26	0.28466

Temp F	Pressure 220 psia Gage pressure 205.3 psig (Sat temp 125.5 F)			Pressure 240 psia Gage pressure 225.3 psig (Sat temp 132.4 F)			Pressure 260 psia Gage pressure 245.3 psig (Sat temp 138.9 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
130	0.2214	109.60	0.19981	—	—	—	—	—	—
140	0.2301	111.79	0.20350	0.2036	110.70	0.20035	0.1807	109.52	0.19720
150	0.2385	113.96	0.20709	0.2119	112.95	0.20407	0.1890	111.87	0.20108
160	0.2466	116.12	0.21060	0.2199	115.18	0.20769	0.1969	114.17	0.20482
170	0.2544	118.26	0.21403	0.2276	117.38	0.21122	0.2045	116.44	0.20846
180	0.2621	120.40	0.21740	0.2349	119.56	0.21465	0.2117	118.68	0.21199
190	0.2696	122.54	0.22072	0.2421	121.74	0.21803	0.2187	120.91	0.21544
200	0.2769	124.67	0.22398	0.2491	123.91	0.22134	0.2255	123.12	0.21882
210	0.2841	126.81	0.22719	0.2560	126.08	0.22461	0.2321	125.33	0.22214
220	0.2911	128.95	0.23035	0.2628	128.25	0.22783	0.2386	127.53	0.22541
230	0.2981	131.09	0.23349	0.2694	130.42	0.23100	0.2450	129.74	0.22863
240	0.3050	133.24	0.23658	0.2759	132.59	0.23413	0.2512	131.94	0.23180
250	0.3118	135.39	0.23964	0.2823	134.77	0.23722	0.2573	134.14	0.23492
260	0.3185	137.56	0.24267	0.2886	136.96	0.24027	0.2633	136.85	0.23801
270	0.3251	139.72	0.24566	0.2948	139.15	0.24329	0.2692	138.56	0.24106
280	0.3316	141.90	0.24862	0.3010	141.34	0.24629	0.2751	140.78	0.24408
290	0.3381	144.09	0.25156	0.3071	143.55	0.24924	0.2809	143.00	0.24707
300	0.3445	146.29	0.25447	0.3131	145.76	0.25217	0.2866	145.23	0.25002
310	0.3509	148.49	0.25736	0.3191	147.98	0.25508	0.2922	147.47	0.25295
320	0.3572	150.71	0.26022	0.3250	150.22	0.25796	0.2978	149.72	0.25585
330	0.3634	152.94	0.26305	0.3309	152.46	0.26082	0.3033	151.97	0.25872
340	0.3696	155.18	0.26587	0.3367	154.71	0.26366	0.3088	154.24	0.26157
350	0.3758	157.43	0.26867	0.3425	156.97	0.26647	0.3142	156.51	0.26440
360	0.3819	159.69	0.27145	0.3482	159.25	0.26926	0.3196	158.80	0.26721
370	0.3880	161.96	0.27420	0.3539	161.53	0.27203	0.3249	161.10	0.26999
380	0.3941	164.25	0.27694	0.3595	163.83	0.27478	0.3302	163.41	0.27275
390	0.4001	166.55	0.27966	0.3652	166.14	0.27752	0.3355	165.73	0.27550
400	0.4061	168.86	0.28237	0.3708	168.46	0.28024	0.3408	168.06	0.27823
410	0.4120	171.18	0.28505	0.3763	170.80	0.28293	0.3460	170.40	0.28094
420	—	—	—	0.3818	173.14	0.28561	0.3512	172.76	0.28364


ETHANE
PRESSURE ENTROPY
DIAGRAM
V=SPECIFIC VOLUME, CU FT PER LB
S=ENTROPY, BTU PER LB -°R
H=ENTHALPY, BTU PER LB
T=TEMPERATURE, F
H=0, S=0, FOR SOLID AT 0°R
SATURATED VAPOR
SATURATED LIQUID
ENTHALPY, BTU PER LB
PRESSURE LB PER SQ IN. ABS
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T=0°
T=-100°
T=100°
T=200°
T=300°
T=400°
T=500°
V=0.20
V=0.50
V=1.00
V=2.00
V=5.00
V=10.0
V=20.0
S=1.50
S=1.60
S=1.70
S=1.80
S=1.90
S=2.00
S=2.10

Chart 5

Table 12. Ethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −148 F Btu per lb		Entropy, datum −148 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−148.0	7.62	14.40*	35.00	14.1	0	219.1	0.000	0.703
−144.4	8.60	12.42*	34.85	12.5	1.9	219.7	0.006	0.697
−140.8	9.67	10.24*	34.70	11.2	3.8	220.3	0.012	0.691
−137.2	10.85	7.84*	34.55	10.1	5.8	220.9	0.018	0.685
−133.6	12.17	5.15*	34.38	9.05	7.7	221.5	0.024	0.680
−130.0	13.60	2.24*	34.22	8.16	9.7	222.1	0.030	0.674
−126.4	15.16	+0.46	34.06	7.41	11.7	222.8	0.036	0.670
−122.8	16.86	2.16	33.90	6.73	13.7	223.6	0.042	0.665
−119.2	18.70	4.00	33.74	6.13	15.8	224.2	0.048	0.660
−115.6	20.69	5.99	33.58	5.62	17.9	224.9	0.054	0.656
−112.0	22.85	8.15	33.40	5.14	19.9	225.6	0.060	0.652
−108.4	25.17	10.47	33.24	4.72	22.1	226.4	0.066	0.648
−104.8	27.67	12.97	33.07	4.33	24.2	227.2	0.072	0.644
−101.2	30.36	15.66	32.90	3.98	26.4	227.9	0.078	0.641
− 97.6	33.23	18.53	32.72	3.66	28.5	228.7	0.084	0.637
− 94.0	36.33	21.63	32.55	3.39	30.7	229.6	0.090	0.634
− 90.4	39.63	24.93	32.38	3.13	33.0	230.5	0.096	0.631
− 86.8	43.14	28.44	32.20	2.90	35.2	231.2	0.102	0.628
− 83.2	46.88	32.18	32.03	2.68	37.5	231.9	0.108	0.624
− 79.6	50.87	36.18	31.86	2.48	39.8	232.8	0.114	0.622
− 76.0	55.09	40.39	31.68	2.30	42.2	233.7	0.120	0.619
− 72.4	59.58	44.88	31.50	2.13	44.5	234.6	0.126	0.617
− 68.8	64.33	49.63	31.32	1.98	46.9	235.5	0.132	0.615
− 65.2	69.35	54.65	31.14	1.84	49.3	236.3	0.139	0.614
− 61.6	74.66	59.96	30.96	1.71	51.8	237.2	0.146	0.612
− 58.0	80.28	65.58	30.76	1.60	54.2	238.2	0.151	0.609
− 54.4	86.15	71.45	30.58	1.48	56.7	239.0	0.157	0.607
− 50.8	92.36	77.66	30.38	1.38	59.2	239.9	0.163	0.605
− 47.2	98.89	84.19	30.18	1.29	61.8	240.9	0.169	0.603
− 43.6	105.7	91.0	29.98	1.21	64.4	241.9	0.175	0.602
− 40.0	112.9	98.2	29.78	1.136	67.0	242.8	0.181	0.600
− 36.4	120.5	105.8	29.58	1.064	69.6	243.7	0.187	0.598
− 32.8	128.4	113.7	29.37	0.996	72.3	244.6	0.194	0.597
− 29.2	136.7	122.0	29.16	0.936	75.0	245.5	0.200	0.596
− 25.8	145.3	130.6	28.94	0.882	77.7	246.4	0.206	0.595
− 22.0	154.4	139.7	28.72	0.831	80.5	247.4	0.212	0.594
− 18.4	163.8	149.1	28.50	0.782	83.4	248.3	0.219	0.593
− 14.8	173.7	159.0	28.27	0.737	86.2	249.1	0.225	0.591
− 11.2	184.0	169.3	28.04	0.695	89.1	249.9	0.231	0.590
− 7.6	194.7	180.0	27.80	0.654	92.0	250.8	0.238	0.589
− 4.0	205.9	191.2	27.57	0.617	95.0	251.6	0.244	0.588
− 0.4	217.6	202.9	27.33	0.582	98.0	252.4	0.250	0.587
+ 3.2	229.7	215.0	27.08	0.549	101.1	253.2	0.257	0.586
6.8	242.3	227.6	26.84	0.518	104.2	254.0	0.263	0.584
10.4	255.4	240.7	26.58	0.490	107.4	254.7	0.270	0.583
14.0	269.1	254.4	26.32	0.463	110.7	255.4	0.277	0.582
17.6	283.2	268.5	26.06	0.437	114.0	256.1	0.283	0.581
21.2	298.0	283.3	25.78	0.412	117.4	256.7	0.290	0.580
24.8	313.3	298.6	25.51	0.388	120.9	257.4	0.297	0.579
28.4	329.2	314.5	25.23	0.366	124.5	258.1	0.304	0.578
32.0	345.7	331.0	24.95	0.346	128.2	258.7	0.312	0.577
35.6	362.7	348.0	24.65	0.328	132.0	259.3	0.319	0.576
39.2	380.5	355.8	24.34	0.311	135.9	259.9	0.327	0.576
42.8	398.8	384.1	24.00	0.296	140.0	260.6	0.335	0.575
46.4	417.8	403.1	23.66	0.282	144.2	261.0	0.343	0.574
50.0	437.5	422.8	23.29	0.267	148.5	261.3	0.351	0.573
53.6	457.9	443.2	22.90	0.251	153.0	262	0.360	0.572
57.2	479.0	464.3	22.48	0.233	157	262	0.368	0.571
60.8	500.9	486.2	22.03	0.213	161	261	0.376	0.568
64.4	523.5	508.8	21.54	0.199	166	261	0.385	0.566
68.0	546.9	532.2	21.00	0.185	170	260	0.393	0.563
71.6	571.1	556.4	20.46	0.171	175	259	0.402	0.560
75.2	596.0	581.3	19.86	0.158	180	257	0.412	0.555
78.8	621.8	607.1	19.16	0.150	185	254	0.421	0.549
82.4	648.4	633.7	18.36	0.133	190	250	0.430	0.541
86.0	675.9	661.1	17.20	0.118	195	244	0.439	0.529
89.6	704.1	689.4	14.40	0.089	210	233	0.467	0.509
− 90.1†	707.8	693.1	12.63	0.079	220	220	0.485	0.485

* Inches of mercury below one standard atmosphere. † Critical.

The Linde Air Products Company, compilers.

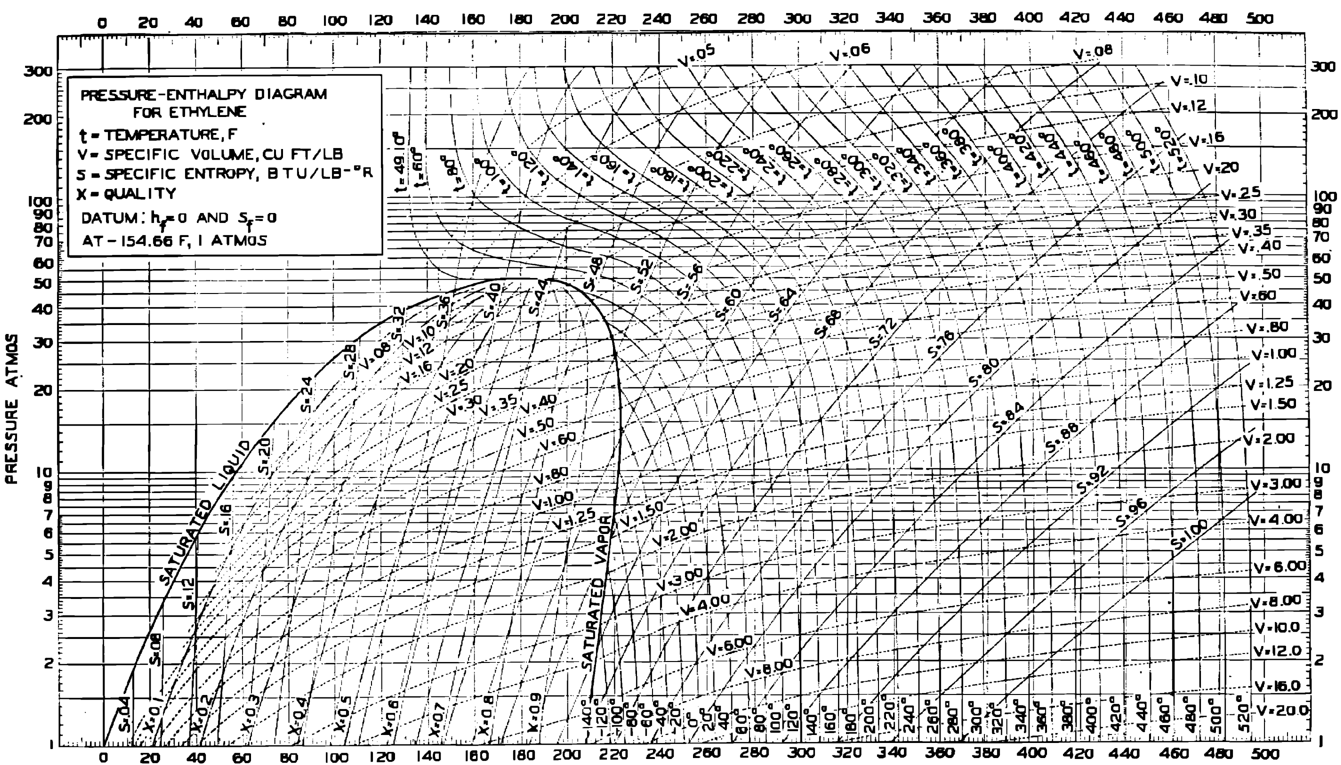

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Chart 6

Table 13. Ethylene—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −176 F Btu per lb		Entropy, datum −176 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−176.8	6.75	16.18*	36.60	16.4	0	213.9	0.000	0.756
−173.2	7.72	14.21*	36.42	14.5	1.8	214.5	0.006	0.749
−169.6	8.80	12.01*	36.23	12.7	3.5	215.1	0.012	0.742
−166.0	10.00	9.57*	36.05	11.1	5.3	215.8	0.018	0.735
−162.4	11.35	7.02*	35.87	9.71	7.1	216.3	0.024	0.728
−158.8	12.80	3.87*	35.69	8.62	9.0	216.9	0.030	0.722
−155.2	14.44	0.52*	35.51	7.52	10.8	217.5	0.036	0.716
−151.6	16.22	1.52	35.32	6.76	12.6	218.1	0.042	0.710
−148.0	18.17	3.47	35.14	6.06	14.5	218.7	0.048	0.704
−144.4	20.36	5.66	34.96	5.49	16.4	219.2	0.054	0.698
−140.8	22.65	7.95	34.78	4.98	18.3	219.9	0.060	0.693
−137.2	25.17	10.47	34.60	4.50	20.2	220.6	0.066	0.688
−133.6	27.91	13.21	34.42	4.08	22.2	221.2	0.072	0.683
−130.0	30.86	16.16	34.23	3.72	24.1	221.8	0.078	0.678
−126.4	34.06	19.36	34.05	3.39	26.0	222.3	0.084	0.673
−122.8	37.50	22.80	33.87	3.11	28.1	223.1	0.090	0.669
−119.2	41.19	26.49	33.69	2.85	30.1	223.8	0.096	0.665
−115.6	45.16	30.46	33.50	2.62	32.2	224.6	0.102	0.661
−112.0	49.40	34.70	33.30	2.40	34.2	225.3	0.108	0.658
−108.4	53.93	39.23	33.11	2.21	36.3	226.0	0.114	0.654
−104.8	58.76	44.06	32.91	2.04	38.4	226.7	0.120	0.650
−101.2	63.90	49.20	32.72	1.89	40.5	227.3	0.125	0.647
− 97.6	69.37	54.67	32.50	1.75	42.6	228.0	0.131	0.643
− 94.0	75.17	60.47	32.28	1.62	44.8	228.7	0.137	0.640
− 90.4	81.32	66.62	32.07	1.51	47.0	229.4	0.143	0.637
− 86.8	87.88	73.18	31.86	1.40	49.2	230.1	0.149	0.634
− 83.2	94.76	80.06	31.64	1.31	51.4	230.8	0.155	0.631
− 79.6	102.0	87.3	31.42	1.22	53.7	231.5	0.161	0.628
− 76.0	109.7	95.0	31.20	1.14	55.9	232.1	0.166	0.626
− 72.4	117.7	103.0	30.98	1.07	58.3	232.9	0.172	0.623
− 68.8	126.2	111.5	30.76	1.00	60.6	233.5	0.178	0.621
− 65.2	135.1	120.4	30.54	0.935	63.0	234.3	0.184	0.618
− 61.6	144.4	129.7	30.31	0.885	65.3	234.9	0.190	0.616
− 58.0	154.2	139.5	30.07	0.826	67.8	235.5	0.196	0.613
− 54.4	164.5	149.8	29.83	0.775	70.3	235.8	0.202	0.610
− 50.8	175.2	160.5	29.58	0.730	72.7	236.1	0.208	0.608
− 47.2	186.7	172.0	29.34	0.685	75.3	236.4	0.214	0.605
− 43.6	198.1	183.4	29.10	0.645	77.8	236.7	0.220	0.602
− 40.0	210.4	196.7	28.84	0.606	80.4	237.0	0.226	0.599
− 36.4	223.2	208.5	28.58	0.571	83.0	237.3	0.232	0.596
− 32.8	236.6	221.9	28.32	0.535	85.7	237.5	0.238	0.594
− 29.2	250.5	235.8	28.04	0.503	88.4	237.4	0.244	0.590
− 25.8	265.2	250.5	27.76	0.474	91.1	237.4	0.250	0.587
− 22.0	280.1	265.4	27.47	0.444	93.9	237.4	0.256	0.584
− 18.4	296.0	281.3	27.18	0.418	96.8	237.2	0.263	0.581
− 14.8	312.2	297.5	26.87	0.392	99.6	236.8	0.269	0.577
− 11.2	329.3	314.6	26.56	0.368	102.7	236.4	0.275	0.573
− 7.6	346.8	332.1	26.24	0.345	105.7	235.9	0.282	0.570
− 4.0	365.5	350.8	25.92	0.323	108.8	235.3	0.288	0.566
− 0.4	384.3	369.6	25.57	0.302	112.0	234.8	0.295	0.562
+ 3.2	404.4	389.7	25.22	0.283	115.2	234.2	0.301	0.558
6.8	424.7	410.0	24.84	0.266	118.7	233.6	0.308	0.555
10.4	446.5	431.8	24.45	0.249	122.3	233.0	0.316	0.551
14.0	468.7	454.0	24.03	0.234	126.0	232.4	0.323	0.548
17.6	491.8	477.1	23.60	0.218	130	232	0.331	0.545
21.2	515.0	500.3	23.13	0.205	134	230	0.339	0.541
24.8	540.4	525.7	22.63	0.190	138	229	0.348	0.537
28.4	556.7	551.0	22.09	0.176	142	228	0.357	0.533
32.0	593.3	578.6	21.48	0.163	147	226	0.367	0.529
35.6	619.8	606.1	20.83	0.149	152	224	0.377	0.523
39.2	649.5	635.1	20.04	0.135	158	222	0.389	0.520
42.8	678.0	664.3	19.12	0.121	166	218	0.404	0.508
46.4	710.3	695.6	17.92	0.103	177	211	0.420	0.494
48.85†	731.8	717.1	13.48	0.074	194	194	0.460	0.460

* Inches of mercury below one standard atmosphere.

† Critical.

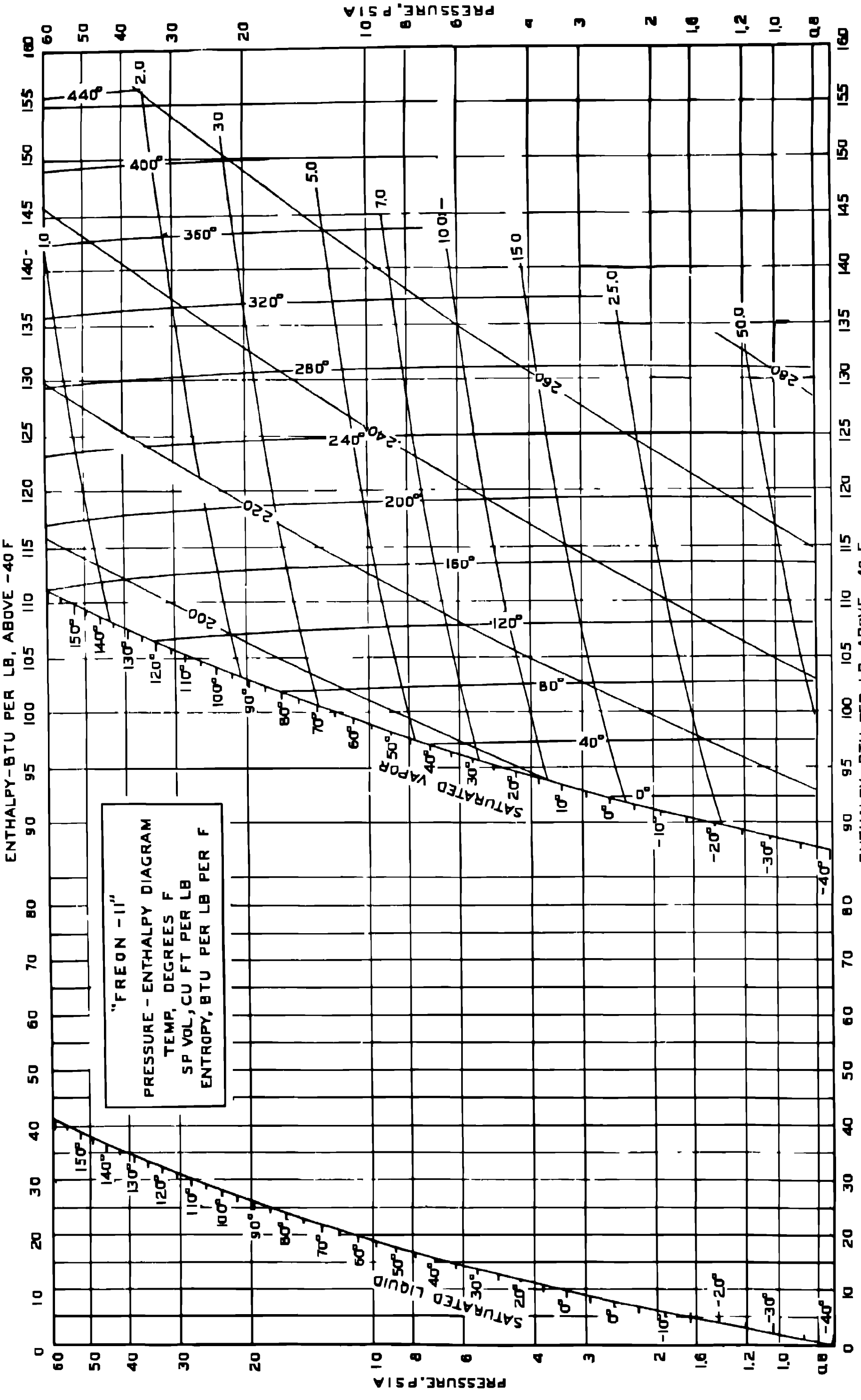
"FREON-11"
PRESSURE-ENTHALPY DIAGRAM
TEMP, DEGREES F
SP VOL, CU FT PER LB
ENTROPY, BTU PER LB PER F
ENTHALPY-BTU PER LB, ABOVE -40 F
PRESSURE, PSIA
0
10
20
30
40
50
60
70
80
90
95
100
105
110
115
120
125
130
135
140
145
150
155
160
60
50
40
30
20
10
8
6
4
3
2
1.6
1.2
1.0
0.8
SATURATED LIQUID
SATURATED VAPOR
150°
140°
130°
120°
110°
100°
90°
80°
70°
60°
50°
40°
30°
20°
10°
0°
-10°
-20°
-30°
-40°
440°
400°
360°
320°
280°
240°
200°
160°
120°
80°
40°
0°
1.0
2.0
3.0
5.0
7.0
10.0
15.0
25.0
50.0
.200
.220
.240
.260
.280
KINETIC CHEMICALS INC

Chart 7

Table 14. F-11 (CCl_3F) Trichloromonofluoromethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−90	0.100	29.72*	0.00964	289.0	−9.7	81.7	−0.0247	0.2226
−80	.157	29.60*	.00961	189.0	−7.7	82.9	− .0195	.2192
−70	.240	29.43*	.00967	127.0	−5.8	84.0	− .0144	.2161
−60	.358	29.19*	.00974	87.0	−3.8	85.2	− .0095	.2132
−50	.520	28.86*	.00981	61.3	−1.9	86.3	− .0047	.2106
−40	0.739	28.42*	0.00988	44.21	0.00	87.48	0.0000	0.2085
−36	0.847	28.20*	.00991	38.93	0.79	87.96	.0019	.2076
−32	0.968	27.95*	.00993	34.37	1.58	88.44	.0037	.2068
−28	1.103	27.67*	.00996	30.44	2.36	88.91	.0055	.2060
−24	1.253	27.37*	.00999	27.03	3.15	89.39	.0073	.2053
−20	1.420	27.03*	0.01002	24.06	3.94	89.87	0.0091	0.2046
−16	1.605	26.65*	.01005	21.47	4.73	90.35	.0109	.2040
−12	1.810	26.24*	.01008	19.20	5.52	90.83	.0127	.2033
− 8	2.035	25.78*	.01011	17.21	6.31	91.31	.0145	.2027
− 4	2.283	25.27*	.01015	15.47	7.10	91.79	.0162	.2021
0	2.555	24.72*	0.01018	13.94	7.89	92.27	0.0179	0.2015
4	2.852	24.11*	.01021	12.58	8.68	92.75	.0197	.2010
5†	2.931	23.95*	.01022	12.27	8.88	92.88	.0201	.2009
8	3.179	23.45*	.01024	11.38	9.48	93.24	.0213	.2005
12	3.534	22.73*	.01027	10.31	10.28	93.72	.0231	.2000
16	3.923	21.94*	.01031	9.359	11.07	94.21	.0248	.1996
20	4.342	21.08*	0.01034	8.519	11.87	94.69	0.0264	0.1991
24	4.801	20.15*	.01037	7.760	12.68	95.18	.0281	.1987
28	5.294	19.14*	.01041	7.087	13.48	95.66	.0297	.1983
32	5.830	18.05*	.01044	6.481	14.28	96.15	.0314	.1979
36	6.411	16.87*	.01048	5.934	15.08	96.63	.0330	.1976
40	7.032	15.61*	0.01051	5.447	15.89	97.11	0.0346	0.1972
44	7.702	14.24*	.01055	5.006	16.70	97.60	.0362	.1969
48	8.422	12.78*	.01058	4.607	17.52	98.08	.0378	.1966
52	9.199	11.20*	.01062	4.245	18.33	98.56	.0394	.1963
56	10.02	9.53*	.01066	3.921	19.15	99.05	.0410	.1960
60	10.90	7.73*	0.01069	3.636	19.96	99.53	0.0426	0.1958
64	11.85	5.80*	.01073	3.356	20.78	100.01	.0442	.1955
68	12.87	3.72*	.01077	3.107	21.61	100.49	.0457	.1953
72	13.95	1.53*	.01081	2.883	22.43	100.97	.0473	.1950
76	15.09	0.39	.01085	2.679	23.26	101.45	.0489	.1948
80	16.31	1.61	0.01088	2.492	24.09	101.93	0.0504	0.1947
84	17.60	2.90	.01092	2.322	24.93	102.41	.0519	.1945
86†	18.28	3.58	.01094	2.242	25.34	102.65	.0527	.1944
88	18.97	4.27	.01096	2.165	25.76	102.89	.0535	.1943
92	20.43	5.73	.01101	2.020	26.60	103.36	.0550	.1941
96	21.97	7.27	.01105	1.887	27.43	103.83	.0565	.1940
100	23.60	8.90	0.01109	1.765	28.27	104.30	0.0580	0.1938
104	25.33	10.63	.01113	1.652	29.12	104.77	.0595	.1937
108	27.15	12.45	.01117	1.548	29.97	105.24	.0610	.1936
112	29.05	14.35	.01122	1.452	30.82	105.71	.0625	.1935
116	31.07	16.37	.01126	1.363	31.67	106.17	.0639	.1934
120	33.20	18.50	0.01130	1.281	32.53	106.63	0.0654	0.1933
124	35.42	20.72	.01135	1.206	33.38	107.09	.0669	.1932
128	37.74	23.04	.01139	1.135	34.24	107.55	.0683	.1931
132	40.23	25.53	.01144	1.068	35.10	108.00	.0698	.1930
136	42.80	28.10	.01149	1.007	35.97	108.46	.0712	.1929
140	45.50	30.80	0.01154	0.9505	36.84	108.91	0.0727	0.1929
144	48.35	33.65	.01159	.8970	37.71	109.35	.0741	.1928
148	51.31	36.61	.01163	.8476	38.59	109.80	.0755	.1927
152	54.41	39.71	.01168	.8014	39.46	110.24	.0770	.1927
156	57.65	42.95	.01173	.7581	40.35	110.69	.0784	.1927
160	61.04	46.34	0.01179	0.7176	41.23	111.12	0.0798	0.1926

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* Inches of mercury below one standard atmosphere.

† Standard cycle temperatures.

Table 15. F-11 (CCl_3F) Trichloromonofluoromethane—Properties of Superheated Vapor

Temp F	Pressure 0.2 psia Gage pressure 29.51 in. vac (Sat temp −74.4 F)			Pressure 0.4 psia Gage pressure 29.11 in. vac (Sat temp −57.2 F)			Pressure 0.6 psia Gage pressure 28.70 in. vac (Sat temp −46.1 F)			Pressure 0:8 psia Gage pressure 28.29 in. vac (Sat temp −37.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	*(150)*	*(83.5)*	*(0.2174)*	*(87.8)*	*(86.5)*	*(0.2125)*	*(53.8)*	*(86.8)*	*(0.2097)*	*(41.09)*	*(87.76)*	*(0.2080)*
0	180	92.5	0.2387	89.8	92.4	0.2285	59.8	92.4	0.2225	44.77	92.37	0.2185
10	183	93.7	.2414	91.7	93.6	.2312	61.1	93.6	.2252	45.74	93.62	.2212
20	187	94.9	.2440	93.7	93.8	.2339	62.4	94.8	.2278	46.73	94.88	.2238
30	191	96.2	.2466	95.6	96.1	.2365	63.7	96.1	.2304	47.70	96.15	.2265
40	195	97.5	.2492	97.6	97.4	.2391	65.0	97.4	.2330	48.69	97.43	.2291
50	199	98.8	0.2518	99.5	98.7	0.2417	66.3	98.7	0.2356	49.66	.98.72	0.2316
60	203	100.1	.2543	102	100.0	.2442	67.7	100.0	.2382	50.65	100.03	.2341
70	207	101.4	.2568	103	101.3	.2467	69.0	101.3	.2407	51.62	101.34	.2366
80	211	102.8	.2593	105	102.7	.2492	70.3	102.7	.2432	52.60	102.67	.2391
90	215	104.1	.2617	107	104.0	.2517	71.6	104.0	.2457	53.38	104.00	.2418
100	219	105.5	0.2641	109	105.4	0.2541	72.9	105.4	0.2481	54.56	105.34	.2440
110	222	106.8	.2665	111	106.7	.2565	74.2	106.7	.2505	55.54	106.69	.2464
120	226	108.1	.2688	113	108.0	.2589	75.5	108.0	.2529	56.62	108.06	.2487
130	230	109.5	.2711	115	109.4	.2612	76.8	109.4	.2552	57.50	109.43	.2511
140	234	110.9	.2734	117	110.8	.2635	78.1	110.8	.2575	58.58	110.81	.2534
150	238	112.3	0.2757	119	112.2	0.2658	79.4	112.2	0.2598	59.46	112.21	0.2557
160	242	113.7	.2780	121	113.6	.2681	80.7	113.6	.2621	60.44	113.61	.2580
170	246	115.1	.2803	123	115.0	.2704	82.0	115.0	.2644	61.42	115.03	.2603
180	250	116.6	.2825	125	116.5	.2726	83.3	116.5	.2667	62.40	116.46	.2625
190	254	118.0	.2847	127	117.9	.2748	84.6	117.9	.2689	63.37	117.89	.2648

Temp F	Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −8.6 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp 16.7 F)			Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp 33.2 F)			Pressure 8.0 psia Gage pressure 13.64 in. vac (Sat temp 45.7 F)		
(*at sat*)	*(17.50)*	*(91.24)*	*(0.2028)*	*(9.194)*	*(94.30)*	*(0.1995)*	*(6.319)*	*(96.29)*	*(0.1978)*	*(4.833)*	*(97.80)*	*(0.1968)*
50	19.81	98.66	0.2183	9.853	98.56	0.2081	6.534	98.46	0.2021	4.875	98.36	0.1979
60	20.20	99.97	.2208	10.05	99.87	.2107	6.668	99.77	.2047	4.976	99.67	.2004
70	20.59	101.28	.2233	10.25	101.18	.2132	6.802	101.09	.2072	5.077	100.99	.2029
80	20.99	102.61	.2258	10.45	102.51	.2157	6.936	102.42	.2097	5.178	102.32	.2054
90	21.38	103.94	.2282	10.65	103.45	.2181	7.069	103.76	.2121	5.279	103.66	.2079
100	21.77	105.28	0.2307	10.84	105.20	0.2205	7.202	105.11	0.2146	5.380	105.02	0.2103
110	22.17	106.64	.2331	11.04	106.56	.2229	7.335	106.47	.2170	5.480	106.38	.2127
120	22.56	108.01	.2354	11.24	107.93	.2253	7.467	107.84	.2194	5.580	107.75	.2151
130	22.95	109.38	.2378	11.44	109.30	.2277	7.600	109.22	.2217	5.680	109.13	.2175
140	23.35	110.76	. 2401	11.64	110.68	.2300	7.733	110.61	.2241	5.780	110.52	.2198
150	23.74	112.17	0.2424	11.83	112.08	0.2323	7.865	112.01	0.2264	5.880	111.92	0.2221
160	24.13	113.57	.2447	12.03	113.49	.2346	7.998	113.41	.2287	5.980	113.34	.2244
170	24.52	114.98	.2470	12.23	114.91	.2369	8.130	114.83	.2309	6.081	114.76	.2267
180	24.92	116.41	.2492	12.43	116.34	.2391	8.263	116.26	.2332	6.181	116.19	.2290
190	25.31	117.85	.2515	12.62	117.77	.2414	8.395	117.70	.2354	6.281	117.63	.2312
200	25.70	119.30	0.2537	12.82	119.22	0.2436	8.527	119.15	0.2376	6.380	119.08	0.2334
210	26.10	120.76	.2559	13.02	120.68	.2458	8.659	120.61	.2398	6.480	120.55	.2356
220	26.49	122.22	.2581	13.22	122.15	.2480	8.791	122.08	.2420	6.579	122.02	.2378
230	26.88	123.70	.2602	13.41	123.63	.2501	8.923	123.56	.2442	6.679	123.50	.2400
240	27.27	125.19	.2624	13.61	125.13	.2523	9.056	125.06	.2463	6.779	124.09	.2421

Temp F	Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp 55.9 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 90.8 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 113.9 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 131.6 F)		
(*at sat*)	*(3.928)*	*(99.04)*	*(0.1960)*	*(2.061)*	*(103.22)*	*(0.1942)*	*(1.409)*	*(105.93)*	*(0.1936)*	*(1.074)*	*(107.96)*	*(0.1930)*
100	4.285	104.93	0.2070	2.099	104.47	0.1964						
110	4.366	106.29	.2094	2.140	105.84	.1989						
120	4.447	107.67	.2118	2.182	107.23	.2013	1.426	106.79	0.1949			
130	4.528	109.05	.2141	2.224	108.63	.2036	1.455	108.19	.1973			
140	4.609	110.44	.2165	2.266	110.03	.2060	1.484	109.60	.1997	1.092	109.17	0.1950
150	4.690	111.85	0.2188	2.307	111.44	0.2083	1.513	111.02	0.2020	1.115	110.60	0.1974
160	4.771	113.26	.2211	2.349	112.87	.2107	1.541	112.45	.2044	1.137	112.04	.1997
170	4.851	114.69	.2234	2.390	114.31	.2130	1.569	113.90	.2067	1.159	113.49	.2021
180	4.932	116.12	.2257	2.431	115.75	.2152	1.598	115.35	.2090	1.181	114.95	.2044
190	5.012	117.56	.2279	2.473	117.19	.2175	1.626	116.81	.2112	1.202	116.42	.2067
200	5.092	119.01	0.2301	2.514	118.65	0.2197	1.654	118.27	0.2135	1.224	117.89	0.2089
210	5.171	120.48	.2323	2.555	120.12	.2219	1.682	119.75	.2157	1.246	119.38	.2112
220	5.251	121.95	.2345	2.596	121.60	.2241	1.711	121.24	.2179	1.268	120.88	.2134
230	5.331	123.43	.2367	2.637	123.09	.2263	1.739	122.74	.2201	1.289	122.39	.2156
240	5.411	124.92	.2388	2.678	124.59	.2285	1.767	125.25	.2223	1.311	124.91	.2178
250	5.491	126.43	0.2409	2.719	126.11	0.2306	1.794	125.77	0.2244	1.332	125.44	0.2199
260	5.571	127.95	.2431	2.760	127.63	.2328	1.822	127.30	.2266	1.353	126.97	.2221
270	5.650	129.48	.2452	2.800	129.16	.2349	1.850	128.84	.2287	1.374	128.52	.2242
280	5.730	131.01	.2473	2.841	130.70	.2370	1.878	130.39	.2308	1.396	130.08	.2263
290	5.809	132.56	.2494	2.881	132.25	.2391	1.905	131.95	.2329	1.417	131.64	.2284

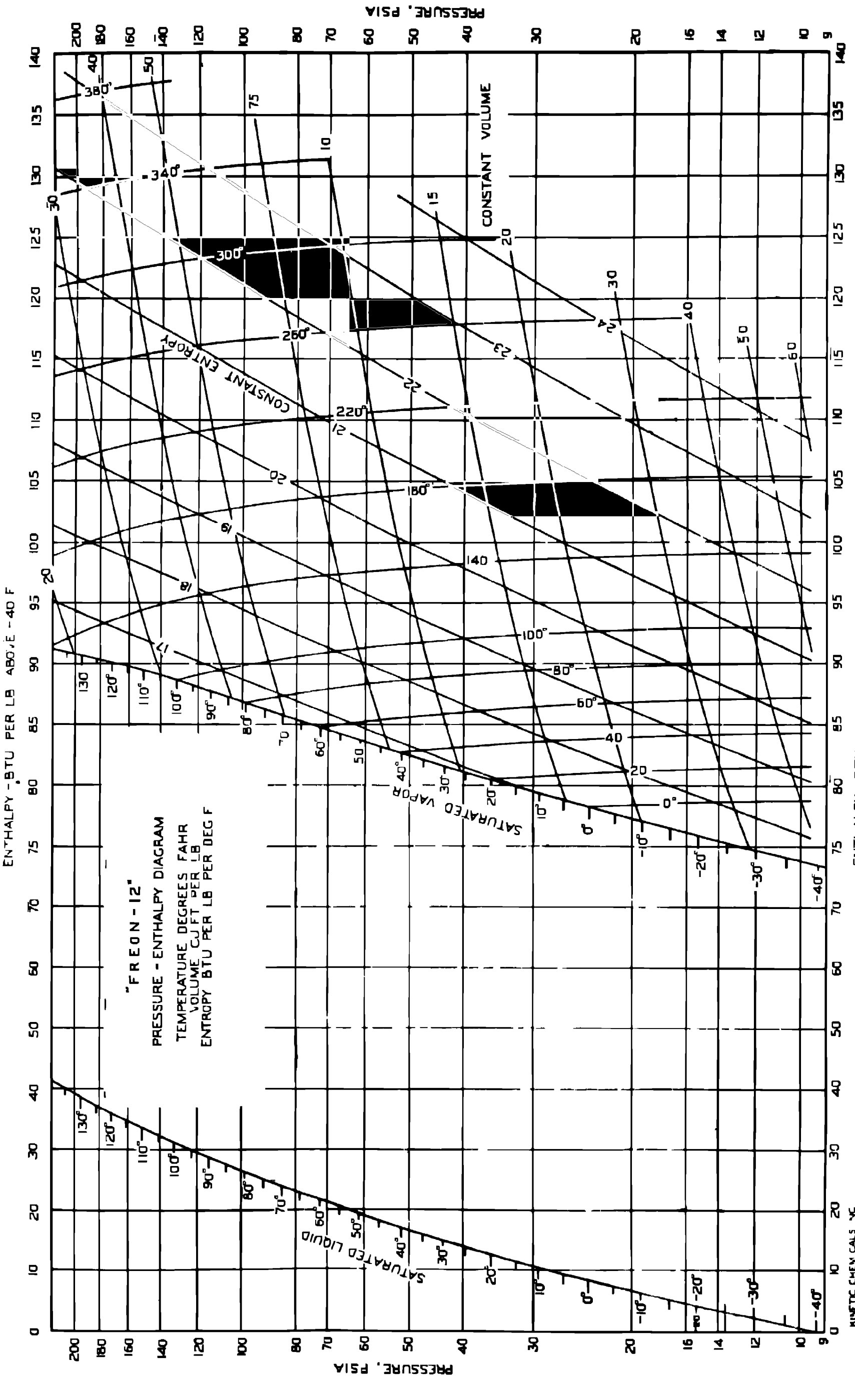
"FREON-12"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE DEGREES FAHR
VOLUME CU FT PER LB
ENTROPY BTU PER LB PER DEG F
PRESSURE, PSIA
200 180 160 140 120 100 80 70 60 50 40 30 20 16 14 12 10 9
ENTHALPY - BTU PER LB ABOVE -40 F
0 10 20 30 40 50 60 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140
CONSTANT VOLUME
CONSTANT ENTROPY
SATURATED VAPOR
SATURATED LIQUID
380° 340° 300° 260° 220° 180° 140 100° 80° 60° 40 20 0°
130° 120° 110° 100° 90° 80° 70° 60° 50° 40° 30° 20° 10° 0° -10° -20° -30° -40°
.40 .50 .75 1.0 1.5 2.0 3.0 4.0 5.0 6.0
.30 .20 .19 .18 .17
.21 .22 .23 .24
KINETIC CHEM CALS INC

Chart 8

Table 16. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−155	0.1163	29.68*	0.00954	232.29	−24.61	60.00	−0.0686	0.2092
−150	0.1527	29.61*	0.00957	179.79	−23.50	60.57	−0.0650	0.2065
−145	.1985	29.52*	.00961	140.52	−22.39	61.14	− .0615	.2040
−140	.2554	29.40*	.00965	110.92	−21.29	61.72	− .0580	.2017
−135	.3256	29.26*	.00969	88.34	−20.19	62.30	− .0546	.1995
−130	.4116	29.08*	.00973	70.94	−19.10	62.88	− .0512	.1975
−125	0.5160	28.87*	0.00977	57.42	−18.02	63.46	−0.0480	0.1955
−120	.6417	28.61*	.00981	46.84	−16.94	64.04	− .0448	.1937
−115	.7921	28.31*	.00985	38.49	−15.85	64.63	− .0416	.1919
−110	.9709	27.94*	.00989	31.84	−14.78	65.22	− .0385	.1903
−105	1.182	27.51*	.00994	26.51	−13.71	65.81	− .0355	.1888
−100	1.430	27.01*	0.00998	22.20	−12.64	66.40	−0.0325	0.1873
− 95	1.719	26.42*	.01003	18.71	−11.58	66.99	− .0295	.1860
− 90	2.054	25.74*	.01007	15.86	−10.51	67.59	− .0266	.1847
− 85	2.441	24.95*	.01012	13.51	− 9.46	68.18	− .0238	.1835
− 80	2.885	24.05*	.01016	11.57	− 8.40	68.77	− .0210	.1823
− 75	3.393	23.01*	0.01021	9.958	− 7.35	69.36	−0.0182	0.1813
− 70	3.971	21.84*	.01026	8.608	− 6.30	69.95	− .0155	.1802
− 65	4.626	20.50*	.01031	7.474	− 5.25	70.54	− .0128	.1793
− 60	5.365	19.00*	.01036	6.516	− 4.20	71.13	− .0102	.1783
− 55	6.195	17.31*	.01041	5.704	− 3.15	71.72	− .0076	.1774
− 50	7.125	15.42*	0.01047	5.012	− 2.11	72.31	−0.0050	0.1767
− 45	8.163	13.31*	.01052	4.420	− 1.06	72.91	− .0025	.1759
− 40	9.317	10.96*	.01057	3.911	0.00	73.50	.0000	.1752
− 35	10.60	8.34*	.01063	3.471	1.01	74.10	+ .0024	.1745
− 30	12.02	5.45*	0.0107	3.088	2.03	74.70	0.00471	0.17387
− 28	12.62	4.23*	.0107	2.950	2.44	74.94	.00565	.17364
− 26	13.26	2.93*	.0107	2.820	2.85	75.18	.00659	.17340
− 24	13.90	1.63*	.0108	2.698	3.25	75.41	.00753	.17317
− 22	14.58	0.24*	.0108	2.583	3.66	75.64	.00846	.17296
− 20	15.28	0.58	0.0108	2.474	4.07	75.87	0.00940	0.17275
− 18	16.01	1.31	.0108	2.370	4.48	76.11	.01033	.17253
− 16	16.77	2.07	.0108	2.271	4.89	76.34	.01126	.17232
− 14	17.55	2.85	.0109	2.177	5.30	76.57	.01218	.17212
− 12	18.37	3.67	.0109	2.088	5.72	76.81	.01310	.17194
− 10	19.20	4.50	0.0109	2.003	6.14	77.05	0.01403	0.17175
− 8	20.08	5.38	.0109	1.922	6.57	77.29	.01496	.17158
− 6	20.98	6.28	.0110	1.845	6.99	77.52	.01589	.17140
− 4	21.91	7.21	.0110	1.772	7.41	77.75	.01682	.17123
− 2	22.87	8.17	.0110	1.703	7.83	77.98	.01775	.17107
0	23.87	9.17	0.0110	1.637	8.25	78.21	0.01869	0.17091
2	24.89	10.19	.0110	1.574	8.67	78.44	.01961	.17075
4	25.96	11.26	.0111	1.514	9.10	78.67	.02052	.17060
5†	26.51	11.81	.0111	1.485	9.32	78.79	.02097	.17052
6	27.05	12.35	.0111	1.457	9.53	78.90	.02143	.17045
8	28.18	13.48	.0111	1.403	9.96	79.13	.02235	.17030
10	29.35	14.65	0.0112	1.351	10.39	79.36	0.02328	0.17015
12	30.56	15.86	.0112	1.301	10.82	79.59	.02419	.17001
14	31.80	17.10	.0112	1.253	11.26	79.82	.02510	.16987
16	33.08	18.38	.0112	1.207	11.70	80.05	.02601	.16974
18	34.40	19.70	.0113	1.163	12.12	80.27	.02692	.16961
20	35.75	21.05	0.0113	1.121	12.55	80.49	0.02783	0.16949
22	37.15	22.45	.0113	1.081	13.00	80.72	.02873	.16938
24	38.58	23.88	.0113	1.043	13.44	80.95	.02963	.16926
26	40.07	25.37	.0114	1.007	13.88	81.17	.03053	.16913
28	41.59	26.89	.0114	0.973	14.32	81.39	.03143	.16900

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* Inches of mercury below one atmosphere.
† Standard cycle temperature.

Table 16. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Liquid and Saturated Vapor (Concluded)

Temp, F	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
30	43.16	28.46	0.0115	0.939	14.76	81.61	0.03233	0.16887
32	44.77	30.07	.0115	.908	15.21	81.83	.03323	.16876
34	46.42	31.72	.0115	.877	15.65	82.05	.03413	.16865
36	48.13	33.43	.0116	.848	16.10	82.27	.03502	.16854
38	49.88	35.18	.0116	.819	16.55	82.49	.03591	.16843
40	51.68	36.98	0.0116	0.792	17.00	82.71	0.03680	0.16833
42	53.51	38.81	.0116	.767	17.46	82.93	.03770	.16823
44	55.40	40.70	.0117	.742	17.91	83.15	.03859	.16813
46	57.35	42.65	.0117	.718	18.36	83.36	.03948	.16803
48	59.35	44.65	.0117	.695	18.82	83.57	.04037	.16794
50	61.39	46.69	0.0118	0.673	19.27	83.78	0.04126	0.16785
52	63.49	48.79	.0118	.652	19.72	83.99	.04215	.16776
54	65.63	50.93	.0118	.632	20.18	84.20	.04304	.16767
56	67.84	53.14	.0119	.612	20.64	84.41	.04392	.16758
58	70.10	55.40	.0119	.593	21.11	84.62	.04480	.16749
60	72.41	57.71	0.0119	0.575	21.57	84.82	0.04568	0.16741
62	74.77	60.07	.0120	.557	22.03	85.02	.04657	.16733
64	77.20	62.50	.0120	.540	22.49	85.22	.04745	.16725
66	79.67	64.97	.0120	.524	22.95	85.42	.04833	.16717
68	82.24	67.54	.0121	.508	23.42	85.62	.04921	.16709
70	84.82	70.12	0.0121	0.493	23.90	85.82	0.05009	0.16701
72	87.50	72.80	.0121	.479	24.37	86.02	.05097	.16693
74	90.20	75.50	.0122	.464	24.84	86.22	.05185	.16685
76	93.00	78.30	.0122	.451	25.32	86.42	.05272	.16677
78	95.85	81.15	.0123	.438	25.80	86.61	.05359	.16669
80	98.76	84.06	0.0123	0.425	26.28	86.80	0.05446	0.16662
82	101.7	87.00	.0123	.413	26.76	86.90	.05534	.16655
84	104.8	90.1	.0124	.401	27.24	87.18	.05621	.16648
86†	107.9	93.2	.0124	.389	27.72	87.37	.05708	.16640
88	111.1	96.4	.0124	.378	28.21	87.56	.05795	.16632
90	114.3	99.6	0.0125	0.368	28.70	87.74	0.05882	0.16624
92	117.7	103.0	.0125	.357	29.19	87.92	.05969	.16616
94	121.0	106.3	.0126	.347	29.68	88.10	.06056	.16608
96	124.5	109.8	.0126	.338	30.18	88.28	.06143	.16600
98	128.0	113.3	.0126	.328	30.67	88.45	.06230	.16592
100	131.6	116.9	0.0127	0.319	31.16	88.62	0.06316	0.16584
102	135.3	120.6	.0127	.310	31.65	88.79	.06403	.16576
104	139.0	124.3	.0128	.302	32.15	88.95	.06490	.16568
106	142.8	128.1	.0128	.293	32.65	89.11	.06577	.16560
108	146.8	132.1	.0129	.285	33.15	89.27	.06663	.16551
110	150.7	136.0	0.0129	0.277	33.65	89.43	0.06749	0.16542
112	154.8	140.1	.0130	.269	34.15	89.58	.06836	.16533
114	158.9	144.2	.0130	.262	34.65	89.73	.06922	.16524
116	163.1	148.4	.0131	.254	35.15	89.87	.07008	.16515
118	167.4	152.7	.0131	.247	35.65	90.01	.07094	.16505
120	171.8	157.1	0.0132	0.240	36.16	90.15	0.07180	0.16495
122	176.2	161.5	.0132	.233	36.66	90.28	.07266	.16484
124	180.8	166.1	.0133	.277	37.16	90.40	.07352	.16473
126	185.4	170.7	.0133	.220	37.67	90.52	.07437	.16462
128	190.1	175.4	.0134	.214	38.18	90.64	.07522	.16450
130	194.9	180.2	0.0134	0.208	38.69	90.76	0.07607	0.16438
132	199.8	185.1	.0135	.202	39.19	90.86	.07691	.16425
134	204.8	190.1	.0135	.196	39.70	90.96	.07775	.16411
136	209.9	195.2	.0136	.191	40.21	91.06	.07858	.16396
138	215.0	200.3	.0137	.185	40.72	91.51	.07941	.16380
140	220.2	205.5	0.0138	0.180	41.24	91.24	0.08024	0.16363

† Standard cycle temperature

Table 17a. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Superheated Vapor

Temp F *t*	Pressure 0.20 psia Gage pressure 29.51 in. vac (Sat temp −144.9 F)			Pressure 0.40 psia Gage pressure 29.11 in. vac (Sat temp −130.7 F)			Pressure 0.60 psia Gage pressure 28.70 in. vac (Sat temp −121.7 F)			Pressure 0.80 psia Gage pressure 28.92 in. vac (Sat temp −114.8 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(139.6)*	*(61.16)*	*(0.2040)*	*(72.84)*	*(62.80)*	*(0.1978)*	*(49.86)*	*(63.86)*	*(0.1943)*	*(38.13)*	*(64.66)*	*(0.1919)*
−140	141.7	61.72	0.2059									
−130	146.1	62.89	.2094	73.00	62.88	0.1980						
−120	150.6	64.07	.2129	75.22	64.06	.2015	50.11	64.04	0.1948			
−110	155.0	65.27	.2164	77.45	65.25	.2050	51.59	65.24	.1983	38.66	65.23	0.1935
−100	159.5	66.47	0.2198	79.67	66.46	0.2084	53.08	66.45	0.2017	39.78	66.44	0.1969
− 90	163.9	67.70	.2231	81.90	67.68	.2117	54.56	67.67	.2050	40.89	67.66	.2003
− 80	168.3	68.93	.2264	84.12	68.92	.2150	56.05	68.91	.2083	41.02	68.90	.2036
− 70	172.8	70.18	.2297	86.34	70.17	.2183	57.53	70.16	.2116	43.12	70.14	.2068
− 60	177.2	71.44	.2329	88.57	71.43	.2215	59.01	71.42	.2148	44.24	71.41	.2100
− 50	181.7	72.71	0.2360	90.79	72.70	0.2246	60.50	72.69	0.2179	45.35	72.68	0.2132
− 40	186.1	74.00	.2391	93.01	73.99	.2277	61.98	73.98	.2210	46.47	73.97	.2163
− 30	190.6	75.30	.2422	95.24	75.29	.2308	63.46	75.28	.2241	47.58	75.27	.2194
− 20	195.0	76.62	.2452	97.46	76.61	.2338	64.95	76.60	.2271	48.69	76.59	.2224
− 10	199.4	77.94	.2482	99.68	77.93	.2368	66.43	77.93	.2301	49.81	77.92	.2254
0	203.9	79.29	0.2511	101.9	79.28	0.2397	67.91	79.27	0.2331	50.92	79.26	0.2283
10	208.3	80.64	.2541	104.1	80.63	.2427	69.39	80.62	.2360	52.03	80.61	.2312
20	212.8	82.01	.2569	106.4	82.00	.2455	70.88	81.99	.2380	53.14	81.98	.2341
30	217.2	83.39	.2598	108.6	83.38	.2484	72.36	83.37	.2417	54.26	83.36	.2370
40	221.6	84.78	.2626	110.8	84.77	.2512	73.84	84.77	.2445	55.37	84.76	.2398
50	226.1	86.19	0.2654	113.0	86.18	0.2540	75.33	86.17	0.2473	56.48	86.17	0.2426
60	230.5	87.61	.2682	115.2	87.60	.2568	76.81	87.59	.2501	57.59	87.59	.2454
70	235.0	89.04	.2709	117.5	89.04	.2595	78.29	89.03	.2528	58.71	89.02	.2481
80	239.4	90.49	.2736	119.7	90.48	.2622	79.77	90.48	.2555	59.82	90.47	.2508
90	243.8	91.95	.2763	121.9	91.94	.2649	81.25	91.94	.2582	60.94	91.93	.2535
100	248.3	93.42	0.2789	124.1	93.42	0.2675	82.73	93.41	0.2609	62.04	93.40	0.2561
110	252.7	94.91	.2816	126.3	94.91	.2702	84.22	94.90	.2635	63.15	94.89	.2588
120	257.2	96.41	.2842	128.6	96.41	.2728	85.70	96.40	.2661	64.27	96.39	.2614
130	261.6	97.93	.2868	130.8	97.92	.2754	87.18	97.91	.2687	65.38	97.91	.2640
140	266.0	99.45	.2893	133.0	99.45	.2779	88.66	99.44	.2713	66.49	99.44	.2665
150	270.5	100.99	0.2919	135.2	100.99	0.2805	90.14	100.98	0.2738	67.60	100.98	0.2691
160	274.9	102.55	.2944	137.5	102.54	.2830	91.62	102.54	.2764	68.71	102.53	.2716
170				139.7	104.11	.2855	93.10	104.10	.2789	69.82	104.10	.2741
180							94.59	105.68	.2814	70.93	105.68	.2766

Temp F	Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −90.8 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp −69.8 F)			Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp −56.2 F)			Pressure 8.0 psia Gage pressure 13.64 in. vac (Sat temp −45.7 F)		
(at sat)	*(16.26)*	*(66.49)*	*(0.1849)*	*(8.649)*	*(69.79)*	*(0.1802)*	*(6.874)*	*(71.68)*	*(0.1776)*	*(4.602)*	*(72.80)*	*(0.1760)*
− 90	16.29	67.59	0.1851									
− 80	16.74	68.83	.1884									
− 70	17.19	70.08	.1917									
− 60	17.64	71.35	.1949	8.772	71.21	0.1833						
− 50	18.09	72.62	0.1980	8.999	72.49	0.1865	5.698	72.37	0.1796			
− 40	18.54	73.91	.2011	9.225	73.78	.1896	6.121	73.67	.1827	4.569	73.56	0.1778
− 30	18.98	75.22	.2042	9.451	75.09	.1927	6.273	74.98	.1858	4.684	74.87	.1809
− 20	19.43	76.53	.2072	9.677	76.41	.1957	6.425	76.30	.1889	4.799	76.20	.1839
− 10	19.88	77.86	.2102	9.903	77.75	.1987	6.577	77.64	.1919	4.914	77.54	.1869
0	20.33	79.21	0.2132	10.13	79.10	0.2017	6.729	78.98	0.1948	5.028	78.89	0.1899
10	20.77	80.56	.2161	10.35	80.46	.2046	6.879	80.34	.1978	5.142	80.26	.1928
20	21.22	81.93	.2190	10.58	81.83	.2075	7.031	81.73	.2007	5.257	81.64	.1957
30	21.67	83.31	.2219	10.80	83.21	.2104	7.182	83.12	.2035	5.370	83.02	.1986
40	22.11	84.71	.2247	11.03	84.61	.2132	7.334	84.52	.2064	5.484	84.43	.2014
50	22.56	86.12	0.2275	11.25	86.03	0.2160	7.485	85.93	0.2092	5.598	85.85	0.2043
60	23.01	87.54	.2302	11.48	87.45	.2187	7.635	87.37	.2119	5.711	87.27	.2070
70	23.45	88.98	.2330	11.70	88.89	.2215	7.786	88.71	.2147	5.824	88.72	.2098
80	23.90	90.43	.2357	11.93	90.35	.2242	7.936	90.26	.2174	5.938	90.18	.2125
90	24.35	91.89	.2384	12.15	91.81	.2269	8.087	91.73	.2201	6.051	91.64	.2152
100	24.79	93.37	0.2410	12.38	93.29	0.2296	8.237	93.21	0.2228	6.165	93.13	0.2179
110	25.24	94.85	.2437	12.60	94.78	.2322	8.388	94.70	.2255	6.278	94.63	.2205
120	25.68	96.36	.2463	12.82	96.28	.2348	8.538	96.20	.2281	6.391	96.13	.2231
130	26.13	97.87	.2489	13.05	97.80	.2374	8.688	97.72	.2307	6.504	97.64	.2257
140	26.58	99.40	.2514	13.27	99.33	.2400	8.838	99.25	.2333	6.617	99.18	.2283
150	27.02	100.94	0.2540	13.50	100.88	0.2425	8.988	100.80	0.2358	6.730	100.73	0.2309
160	24.47	102.50	.2565	13.72	102.43	.2451	9.138	102.36	.2383	6.843	102.29	.2334
170	27.91	104.06	.2590	13.94	104.00	.2476	9.288	103.93	.2408	6.955	103.87	.2359
180	28.36	105.65	.2615	14.17	105.58	.2501	9.437	105.51	.2433	7.068	105.44	.2384
190	28.80	107.24	.2640	14.39	107.18	.2526	9.587	107.11	.2458	7.181	107.05	.2409
200	29.25	108.85	0.2665	14.62	108.79	0.2550	9.737	108.72	0.2483	7.294	108.67	0.2434
210	29.70	110.47	.2689	14.84	110.41	.2575	9.886	110.34	.2507	7.407	110.28	.2458
220				15.06	112.05	.2599	10.04	111.98	.2532	7.520	111.93	.2483
230				15.29	113.70	.2623	10.19	113.63	.2556	7.633	113.57	.2507
240				15.51	115.36	.2647	10.34	115.29	.2580	7.746	115.23	.2531

Table 17b. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Superheated Vapor

Temp F *t*	Pressure 10 psia Gage pressure 9.6 in. vac (Sat temp −37.3 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp −8.2 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 11.1 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 25.9 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*3.652*)	(*75.80*)	(*0.17480*)	(*1.926*)	(*77.27*)	(*0.17160*)	(*1.323*)	(*79.47*)	(*0.17008*)	(*1.009*)	(*81.16*)	(*0.16914*)
−30	3.728	74.77	0.17704									
−20	3.821	76.11	.18008									
−10	3.913	77.46	.18310									
0	4.006	78.81	0.18611	1.965	78.39	0.17407						
10	4.098	80.18	.18905	2.013	79.76	.17704						
20	4.189	81.56	.19194	2.060	81.14	.17996	1.350	80.73	0.17269			
30	4.280	82.94	.19482	2.107	82.55	.18286	1.383	82.15	.17562	1.019	81.76	0.17030
40	4.371	84.35	.19766	2.155	83.97	.18573	1.415	83.58	.17851	1.044	83.20	.17322
50	4.463	85.77	0.20047	2.203	85.40	0.18858	1.448	85.03	0.18138	1.070	84.65	0.17612
60	4.556	87.19	.20326	2.250	86.85	.19138	1.480	86.48	.18420	1.095	86.11	.17896
70	4.648	88.64	.20601	2.297	88.31	.19415	1.512	87.95	.18699	1.120	87.60	.18178
80	4.740	90.11	.20874	2.343	89.78	.19688	1.544	89.43	.18974	1.144	89.09	.18455
90	4.832	91.58	.21144	2.390	91.26	.19959	1.576	90.91	.19249	1.169	90.58	.18731
100	4.923	93.05	0.21411	2.437	92.75	0.20229	1.608	92.41	0.19519	1.194	92.09	0.19004
110	5.015	94.56	.21676	2.483	94.26	.20494	1.640	93.93	.19787	1.218	93.62	.19272
120	5.107	96.07	.21940	2.530	95.78	.20759	1.672	95.46	.20053	1.242	95.15	.19538
130	5.198	97.59	.22199	2.577	97.31	.21020	1.703	97.00	.20315	1.267	96.70	.19803
140	5.289	99.14	.22458	2.623	98.85	.21280	1.735	98.54	.20577	1.291	98.26	.20066
150	5.379	100.66	0.22713	2.699	100.40	0.21537	1.767	100.11	0.20836	1.315	99.83	0.20325
160	5.470	102.24	.22967	2.716	101.97	.21792	1.799	101.69	.21092	1.340	101.42	.20583
170	5.560	103.81	.23218	2.762	103.56	.22045	1.829	103.28	.21344	1.364	103.02	.20838
180	5.650	105.40	.23469	2.808	105.15	.22297	1.860	104.88	.21597	1.388	104.63	.21092
190	5.740	107.00	.21717	2.854	106.76	.22545	1.891	106.49	.21846	1.412	106.25	.21343
200	5.831	108.63	0.23963	2.901	108.38	0.22794	1.923	108.12	0.22096	1.435	107.88	0.21592
210	5.921	110.25	.24208	2.947	110.01	.23039	1.954	109.76	.22342	1.459	109.52	.21840
220	6.011	111.88	.24451	2.992	111.65	.23283	1.986	111.41	.22588	1.482	111.17	.22085
230	6.101	113.53	.24692	3.038	113.31	.23524	2.017	113.08	.22830	1.506	112.84	.22329
240				3.084	114.98	.23766	2.048	114.75	.23072	1.530	114.52	.22572
250				3.130	116.67	0.24005	2.079	116.44	0.23312	1.554	116.21	0.22813
260				3.117	118.36	.24242	2.110	118.15	.23550	1.577	117.92	.23052
270				3.223	120.07	.24477	2.141	119.87	.23787	1.601	119.65	.23289
280							2.172	121.60	.24023	1.625	121.40	.23526
290										1.649	123.15	.23760
300										1.673	124.92	0.23994

Temp F *t*	Pressure 50 psia Gage pressure 35.3 psig (Sat temp 38.3 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 48.7 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 57.9 F)			Pressure 80 psia Gage pressure 65.3 psig (Sat temp 66.3 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.817*)	(*82.52*)	(*0.16841*)	(*0.688*)	(*83.65*)	(*0.16791*)	(*0.594*)	(*84.61*)	(*0.16749*)	(*0.521*)	(*85.46*)	(*0.16716*)
40	0.821	82.76	0.16895									
50	0.842	84.24	0.17187	0.690	83.83	0.16829						
60	0.863	85.72	.17475	0.708	85.33	.17120	0.597	84.94	0.16310			
70	0.884	87.22	.17760	0.726	86.84	.17407	0.612	86.44	.17097	0.526	86.01	0.16819
80	0.904	88.72	.18040	0.743	88.35	.17689	0.628	87.96	.17382	0.540	87.56	.17108
90	0.924	90.23	.18317	0.760	89.87	.17968	0.643	89.49	.17665	0.554	89.12	.17394
100	0.944	91.75	0.18591	0.778	91.41	0.18246	0.658	91.03	0.17943	0.568	90.68	0.17675
110	0.964	93.29	.18862	0.795	92.96	.18519	0.673	92.59	.18219	0.582	92.26	.17954
120	0.984	94.83	.19132	0.812	94.51	.18789	0.689	94.16	.18493	0.596	93.84	.18229
130	1.004	96.39	.19397	0.829	96.07	.19056	0.704	95.75	.18763	0.609	95.43	.18500
140	1.024	97.96	.19662	0.846	97.65	.19323	0.719	97.34	.19030	0.623	97.03	.18771
150	1.044	99.54	0.19923	0.863	99.24	0.19585	0.733	98.94	0.19293	0.636	98.64	0.19035
160	1.064	101.14	.20182	0.880	100.84	.19846	0.748	100.54	.19555	0.649	100.26	.19298
170	1.084	102.75	.20439	0.897	102.45	.20104	0.763	102.16	.19814	0.662	101.88	.19558
180	1.103	104.36	.20694	0.913	104.07	.20360	0.777	103.80	.20071	0.675	103.52	.19817
190	1.123	105.98	.20946	0.930	105.71	.20613	0.792	105.45	.20325	0.688	105.18	.20073
200	1.142	107.62	0.21196	0.946	107.36	0.20865	0.806	107.10	0.20579	0.701	106.84	0.20328
210	1.162	109.28	.21444	0.962	109.02	.21113	0.820	108.76	.20829	0.714	108.51	.20580
220	1.181	110.95	.21691	0.979	110.69	.21361	0.835	110.43	.21079	0.726	110.19	.20828
230	1.200	112.62	.21935	0.995	112.37	.21607	0.849	112.13	.21325	0.739	111.88	.21076
240	1.220	114.31	.22179	1.012	114.06	.21853	0.863	113.83	.21570	0.751	113.58	.21321
250	1.239	116.00	0.22419	1.028	115.77	0.22094	0.878	115.55	0.21815	0.764	115.30	0.21566
260	1.258	117.71	.22660	1.044	117.49	.22334	0.892	117.28	.22057	0.777	117.03	.21809
270	1.277	119.44	.22898	1.060	119.23	.22573	0.906	119.02	.22296	0.789	118.78	.22049
280	1.296	121.18	.23134	1.076	120.97	.22810	0.920	120.76	.22534	0.802	120.54	.22289
290	1.314	122.93	.23367	1.092	122.73	.23045	0.934	122.52	.22770	0.814	122.30	.22525
300	1.332	124.69	0.23600	1.108	124.50	0.23280	0.948	124.29	0.23006	0.826	124.08	0.22760
310	1.350	126.45	.23831	1.124	126.28	.23513	0.961	126.07	.23239	0.839	125.88	.22995
320				1.140	128.07	.23745	0.975	127.88	.23471	0.851	127.70	.23229
330							0.989	129.70	.23702	0.864	129.52	.23461

Table 17c. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Superheated Vapor

Temp F	Pressure 90 psia Gage pressure 75.3 psig (Sat temp 73.9 F)			Pressure 100 psia Gage pressure 85.3 psig (Sat temp 80.0 F)			Pressure 110 psia Gage pressure 95.3 psig (Sat temp 87.3 F)			Pressure 120 psia Gage pressure 105.3 psig (Sat temp 93.4 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(*at sat*)	(*0.465*)	(*86.21*)	(*0.16685*)	(*0.419*)	(*86.89*)	(*0.16659*)	(*0.382*)	(*87.60*)	(*0.16636*)	(*0.350*)	(*88.06*)	(*0.16610*)
80	0.473	87.17	0.16862									
90	0.486	88.74	.17149	0.430	88.32	0.16926	0.385	87.91	0.16711			
100	0.499	90.31	0.17433	0.442	89.93	0.17210	0.396	89.51	0.17001	0.357	89.13	0.16803
110	0.511	91.89	.17713	0.454	91.54	.17493	0.407	91.12	.17287	0.367	90.75	.17090
120	0.523	93.48	.17990	0.465	93.15	.17733	0.417	92.74	.17568	0.377	92.38	.17374
130	0.535	95.08	.18262	0.477	94.76	.18049	0.428	94.37	.17845	0.387	94.01	.17654
140	0.547	96.69	.18533	0.488	96.37	.18231	0.438	96.01	.18122	0.397	95.65	.17932
150	0.559	98.31	0.18799	0.499	97.99	0.18590	0.449	97.66	0.18394	0.407	97.30	0.18207
160	0.571	100.04	.19065	0.510	99.63	.18856	0.459	99.31	.18660	0.417	98.96	.18474
170	0.584	101.58	.19327	0.521	101.28	.19120	0.469	100.97	.18924	0.426	100.63	.18743
180	0.596	103.23	.19588	0.531	102.84	.19381	0.479	102.64	.19187	0.436	102.31	.19011
190	0.607	104.89	.19845	0.542	104.61	.19638	0.489	104.32	.19447	0.445	104.00	.19271
200	0.619	106.56	0.20101	0.553	106.20	0.19894	0.499	106.01	0.19706	0.454	105.70	0.19529
210	0.630	108.24	.20353	0.563	107.98	.20148	0.509	107.71	.19962	0.463	107.41	.19785
220	0.642	109.93	.20603	0.574	109.68	.20401	0.519	109.42	.20216	0.472	109.13	.20041
230	0.653	111.63	.20852	0.585	111.39	.20650	0.528	111.14	.20464	0.482	110.86	.20294
240	0.665	113.35	.21100	0.595	113.11	.20899	0.538	112.87	.20712	0.491	112.60	.20545
250	0.676	115.08	0.21345	0.606	114.84	0.21145	0.548	114.61	0.20959	0.500	114.35	0.20792
260	0.688	116.82	.21589	0.616	116.58	.21389	0.557	116.36	.21205	0.508	116.11	.21035
270	0.699	118.57	.21831	0.626	118.33	.21631	0.567	118.12	.21448	0.517	117.88	.21279
280	0.710	120.33	.22070	0.636	120.10	.21870	0.576	119.89	.21690	0.526	119.66	.21521
290	0.721	122.10	.22306	0.646	121.88	.22108	0.586	121.68	.21930	0.534	121.45	.21760
300	0.732	123.88	0.22542	0.657	123.67	0.22347	0.595	123.48	0.22167	0.543	123.25	0.22000
310	0.743	125.97	.22776	0.667	125.47	.22583	0.605	125.29	.22405	0.552	125.07	.22238
320	0.754	127.48	.23008	0.677	127.28	.22817	0.614	127.11	.22639	0.560	126.90	.22472
330	0.765	129.31	.23240	0.687	129.10	.23050	0.623	128.94	.22872	0.569	128.74	.22707
340				0.697	130.94	.23281	0.632	130.78	.23103	0.578	130.59	.22940
350				0.707	132.80	0.23150	0.641	132.63	0.23333	0.586	132.45	0.23171
360				0.718	134.68	.23738	6.651	134.50	.23562	0.595	134.32	.23400
370												

Temp F	Pressure 140 psia Gage pressure 125.3 psig (Sat temp 104.5 F)			Pressure 160 psia Gage pressure 145.3 psig (Sat temp 114.5 F)			Pressure 180 psia Gage pressure 165.3 psig (Sat temp 123.7 F)			Pressure 200 psia Gage pressure 185.3 psig (Sat temp 132.1 F)		
(*at sat*)	(*0.298*)	(*88.89*)	(*0.16566*)	(*0.259*)	(*89.77*)	(*0.16522*)	(*0.228*)	(*90.38*)	(*0.16476*)	(*0.202*)	(*90.86*)	(*0.16424*)
110	0.304	89.92	0.16725									
120	0.314	91.60	.17021	0.264	90.68	0.16682						
130	0.323	93.28	.17306	0.273	92.40	.16977	0.233	91.47	0.16665			
140	0.332	94.96	.17590	0.282	94.12	.17269	0.241	93.23	.16964	0.208	92.30	0.16611
150	0.341	96.65	0.17868	0.290	95.84	0.17553	0.249	94.99	0.17254	0.216	94.10	0.16966
160	0.350	98.34	.18142	0.298	97.57	.17832	0.257	96.75	.17541	0.224	95.90	.17262
170	0.358	100.03	.18412	0.306	99.31	.18106	0.265	98.52	.17823	0.231	97.70	.17551
180	0.366	101.72	.18678	0.313	101.05	.18377	0.272	100.29	.18102	0.238	99.51	.17838
190	0.374	103.42	.18941	0.321	102.80	.18646	0.280	102.07	.18377	0.245	101.32	.18115
200	0.383	105.14	0.19205	0.329	104.55	0.18913	0.287	103.85	0.18648	0.252	103.13	0.18388
210	0.391	106.86	.19466	0.336	106.31	.19175	0.294	105.63	.18912	0.258	104.94	.18659
220	0.399	108.59	.19724	0.344	108.07	.19435	0.301	107.42	.19174	0.265	106.76	.18927
230	0.407	110.33	.19976	0.351	109.83	.19693	0.307	109.21	.19433	0.272	108.58	.19192
240	0.415	112.09	.20229	0.358	111.60	.19949	0.314	111.01	.19693	0.278	110.40	.19455
250	0.423	113.85	0.20479	0.366	113.38	0.20203	0.321	112.81	0.19947	0.284	112.23	0.19713
260	0.431	115.63	.20728	0.373	115.17	.20453	0.327	114.62	.20199	0.290	114.06	.19967
270	0.439	117.42	.20974	0.380	116.97	.20700	0.334	116.44	.20449	0.296	115.89	.20217
280	0.447	119.32	.21219	0.387	118.78	.20946	0.340	118.26	.20698	0.302	117.73	.20467
290	0.455	121.03	.21461	0.394	120.60	.21189	0.347	120.09	.20944	0.308	119.58	.20715
300	0.462	122.85	0.21701	0.401	122.43	0.21432	0.353	121.92	0.21187	0.314	121.44	0.20961
310	0.470	124.67	.21939	0.408	124.27	.21672	0.359	123.76	.21428	0.320	123.31	.21204
320	0.477	126.50	.22174	0.414	126.12	.21909	0.365	125.61	.21665	0.326	125.19	.21445
330	0.485	128.33	.22411	0.421	127.98	.22145	0.371	127.47	.21904	0.331	127.08	.21685
340	0.492	130.17	.22646	0.428	129.85	.22381	0.377	129.34	.22140	0.337	128.98	.21923
350	0.500	132.02	0.22880	0.435	131.73	0.22616	0.383	131.23	0.22374	0.343	130.89	0.22159
360	0.507	133.89	.23109	0.442	133.63	.22849	0.390	133.13	.22608	0.348	132.81	.22392
370	0.515	135.78	.23336	0.448	135.55	.23079	0.396	135.05	.22840	0.354	134.74	.22624
380							0.402	136.98	.23072	0.360	136.68	.22856
390							0.408	138.91	.23301	0.365	138.63	.23085
400							0.414	140.85	0.23529	0.370	140.59	0.23314

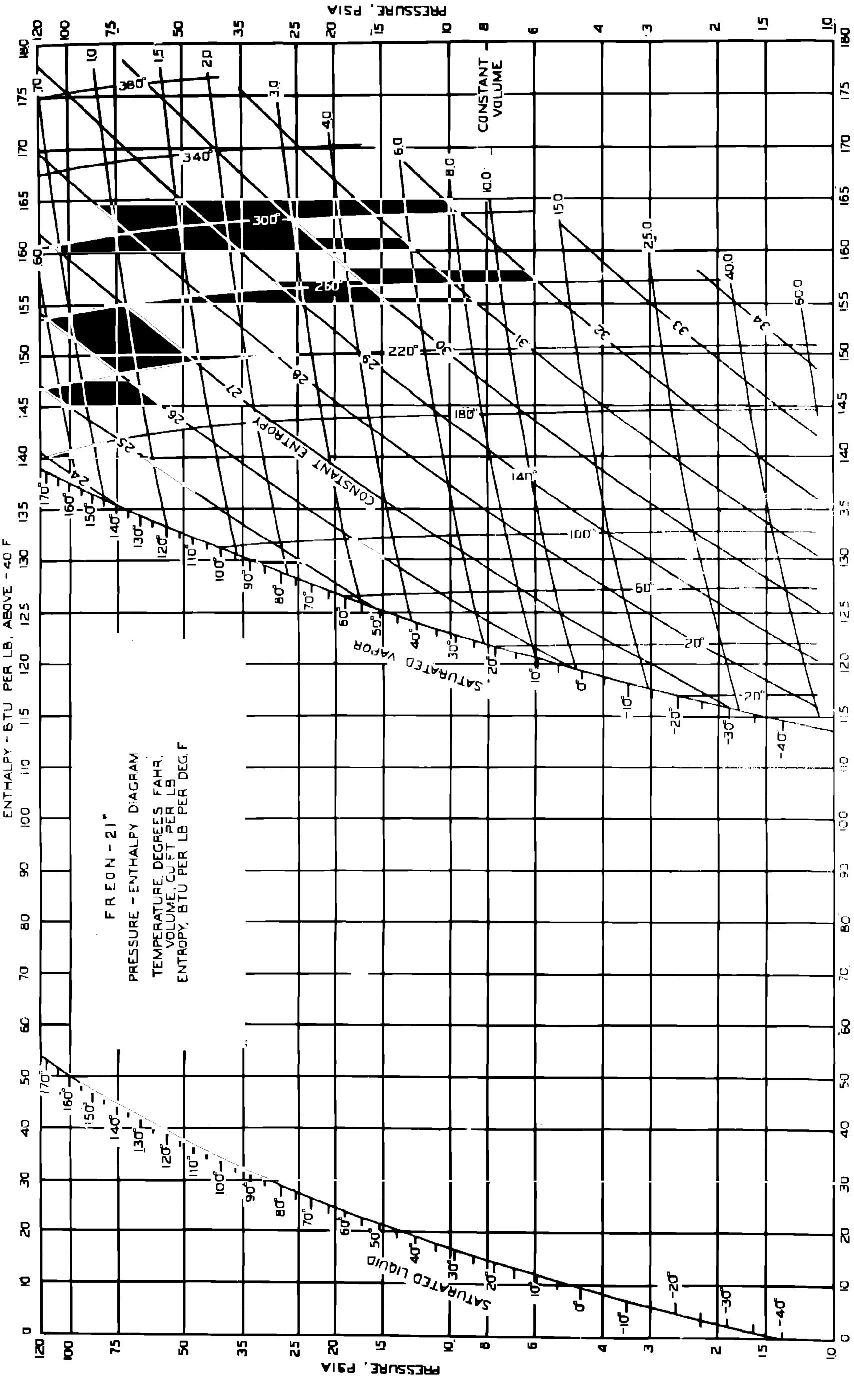

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Chart 9

Table 18. F-21 ($CHCl_2F$) Dichloromonofluoromethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−40	1.358	27.16*	0.01058	32.09	0.00	114.56	0.0000	0.2730
−30	1.888	26.08*	.01066	23.61	2.36	115.76	.0055	.2695
−20	2.578	24.67*	.01075	17.66	4.71	116.96	.0109	.2663
−10	3.463	22.87*	.01084	13.43	7.07	118.17	.0162	.2633
0	4.582	20.59*	.01093	10.35	9.44	119.37	.0214	.2606
2	4.838	20.07*	0.01095	9.840	9.92	119.61	0.0225	0.2601
4	5.105	19.53*	.01096	9.361	10.39	119.85	.0235	.2596
5†	5.243	19.25*	.01097	9.132	10.63	119.97	.0240	.2593
6	5.384	18.96*	.01098	8.910	10.86	120.09	.0245	.2591
8	5.674	18.37*	.01100	8.486	11.33	120.33	.0255	.2586
10	5.978	17.75*	0.01102	8.085	11.81	120.57	0.0265	0.2581
12	6.294	17.11*	.01104	7.707	12.29	120.81	.0275	.2577
14	6.625	16.43*	.01106	7.349	12.77	121.05	.0286	.2572
16	6.968	15.73*	.01108	7.012	13.25	121.30	.0296	.2567
18	7.325	15.01*	.01110	6.694	13.73	121.54	.0306	.2563
20	7.699	14.25*	0.01112	6.392	14.21	121.78	0.0316	0.2559
22	8.087	13.46*	.01114	6.107	14.68	122.02	.0326	.2555
24	8.488	12.64*	.01116	5.838	15.16	122.26	.0336	.2550
26	8.906	11.79*	.01118	5.584	15.64	122.50	.0346	.2546
28	9.341	10.90*	.01120	5.342	16.12	122.74	.0356	.2542
30	9.793	9.98*	0.01122	5.112	16.61	122.98	0.0365	0.2538
32	10.26	9.03*	.01124	4.894	17.09	123.22	.0375	.2534
34	10.75	8.03*	.01126	4.688	17.58	123.46	.0385	.2530
36	11.26	7.00*	.01128	4.492	18.07	123.71	.0395	.2526
38	11.78	5.94*	.01130	4.306	18.55	123.95	.0405	.2523
40	12.32	4.84*	0.01132	4.130	19.04	124.19	0.0414	0.2519
42	12.88	3.70*	.01134	3.963	19.53	124.43	.0424	.2516
44	13.46	2.52*	.01136	3.804	20.02	124.67	.0434	.2512
46	14.07	1.28*	.01138	3.651	20.51	124.91	.0444	.2509
48	14.69	0.01*	.01140	3.507	21.00	125.15	.0453	.2505
50	15.33	0.63	0.01142	3.370	21.49	125.39	0.0463	0.2502
52	16.00	1.30	.01145	3.239	21.99	125.63	.0473	.2499
54	16.69	1.99	.01147	3.114	22.48	125.87	.0482	.2495
56	17.40	2.70	.01149	2.995	22.98	126.11	.0492	.2492
58	18.14	3.44	.01151	2.881	23.48	126.36	.0502	.2489
60	18.90	4.20	0.01153	2.773	23.98	126.60	0.0511	0.2486
62	19.69	4.99	.01155	2.669	24.49	126.84	.0521	.2483
64	20.50	5.80	.01158	2.570	24.99	127.08	.0530	.2480
66	21.34	6.64	.01160	2.476	25.49	127.32	.0540	.2477
68	22.20	7.50	.01162	2.386	25.99	127.56	.0549	.2474
70	23.08	8.38	0.01164	2.300	26.49	127.79	0.0559	0.2471
72	24.00	9.30	.01167	2.218	27.00	128.03	.0568	.2469
74	24.95	10.25	.01169	2.139	27.51	128.27	.0577	.2466
76	25.92	11.22	.01171	2.064	28.02	128.51	.0587	.2463
78	26.93	12.23	.01174	1.992	28.52	128.74	.0596	.2460
80	27.96	13.26	0.01176	1.923	29.03	128.98	0.0606	0.2458
82	29.02	14.32	.01178	1.857	29.54	129.21	.0615	.2455
84	30.11	15.41	.01181	1.794	30.05	129.44	.0624	.2453
86†	31.23	16.53	.01183	1.733	30.56	129.68	.0634	.2450
88	32.39	17.69	.01186	1.675	31.07	129.91	.0643	.2448
90	33.58	18.88	0.01188	1.619	31.59	130.14	0.0652	0.2446
92	34.80	20.10	.01190	1.565	32.11	130.38	.0662	.2443
94	36.06	21.36	.01193	1.513	32.63	130.61	.0671	.2441
96	37.35	22.65	.01195	1.464	33.15	130.84	.0680	.2439
98	38.68	23.98	.01198	1.417	33.67	131.07	.0690	.2437
100	40.04	25.34	0.01200	1.371	34.18	131.29	0.0699	0.2434
110	47.40	32.70	.01213	1.169	36.79	132.42	.0745	.2424
120	55.75	41.05	.01226	1.001	39.46	133.53	.0791	.2414
130	65.15	50.45	.01240	.8623	42.13	134.61	.0837	.2405
140	75.72	61.02	.01254	.7457	44.86	135.66	.0882	.2396
150	87.51	72.81	0.01269	0.6476	47.62	136.68	0.0927	0.2388
160	100.6	85.91	.01284	.5646	50.43	137.69	.0972	.2381

* Inches of mercury below one atmosphere.
† Standard cycle temperatures.

Table 19. F-21 ($CHCl_2F$) Dichloromonofluoromethane—Properties of Superheated Vapor

Temp F	Pressure 2 psia Gage pressure 25.85 in. vac (Sat temp −28.2 F)			Pressure 4 psia Gage pressure 21.78 in. vac (Sat temp 4.9 F)			Pressure 6 psia Gage pressure 17.71 in. vac (Sat temp 10.1 F)			Pressure 8 psia Gage pressure 13.64 in. vac (Sat temp 21.6 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(22.37)*	*(116.98)*	*(0.2689)*	*(11.74)*	*(118.78)*	*(0.2619)*	*(8.066)*	*(120.68)*	*(0.2581)*	*(6.169)*	*(121.96)*	*(0.2556)*
0	23.85	119.49	0.2768	11.87	119.40	0.2633	—	—	—	—	—	—
10	24.38	120.76	.2795	12.14	120.66	.2660	—	—	—	—	—	—
20	24.90	122.04	.2882	12.40	121.95	.2687	8.232	121.86	0.2608	—	—	—
30	25.43	123.34	.2849	12.66	123.25	.2714	8.408	123.16	.2635	6.281	123.06	0.2578
40	25.95	124.65	.2876	12.93	124.57	.2741	8.585	124.48	.2661	6.414	124.39	.2605
50	26.47	125.98	0.2902	13.19	125.90	0.2767	8.762	125.81	0.2688	6.547	125.73	0.2631
60	27.00	127.33	.2928	13.45	127.25	.2793	8.939	127.17	.2714	6.681	127.08	.2657
70	27.52	128.69	.2954	13.72	128.61	.2819	9.115	128.53	.2740	6.814	128.44	.2683
80	28.05	130.07	.2979	13.98	129.99	.2845	9.291	129.91	.2766	6.947	129.82	.2709
90	28.57	131.45	.3005	14.24	131.37	.2870	9.467	131.29	.2791	7.079	131.21	.2735
100	29.09	132.85	0.3030	14.51	132.77	0.2895	9.643	132.69	0.2816	7.212	132.61	0.2760
110	29.62	134.26	.3055	14.77	134.18	.2920	9.819	134.11	.2841	7.345	134.03	.2785
120	30.14	135.68	.3080	15.03	135.61	.2945	9.995	135.54	.2866	7.477	135.46	.2810
130	30.66	137.12	.3104	15.29	137.04	.2970	10.17	136.97	.2891	7.609	136.90	.2835
140	31.19	138.57	.3129	15.56	138.49	.2994	10.35	138.42	.2915	7.742	138.35	.2859
150	31.71	140.03	0.3153	15.82	139.96	0.3019	10.52	139.89	0.2940	7.874	139.82	0.2883
160	32.23	141.52	.3177	16.08	141.45	.3043	10.70	141.39	.2964	8.006	141.31	.2908
170	32.76	143.02	.3201	16.35	142.95	.3067	10.87	142.89	.2988	8.138	142.82	.2932
180	33.28	144.53	.3225	16.61	144.46	.3091	11.05	144.40	.3012	8.270	144.33	.2956
190	33.80	146.05	.3249	16.87	145.98	.3114	11.22	145.92	.3035	8.402	145.86	.2979

Temp F	Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp 30.9 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 62.8 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 83.8 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 99.9 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(5.014)*	*(123.10)*	*(0.2536)*	*(2.630)*	*(126.94)*	*(0.2482)*	*(1.800)*	*(129.42)*	*(0.2453)*	*(1.373)*	*(131.28)*	*(0.2436)*
100	5.753	132.54	0.2176	2.834	132.14	0.2578	1.860	131.72	0.2495	1.373	131.29	0.2435
110	5.860	133.95	.2741	2.889	133.56	.2603	1.898	133.16	.2520	1.402	132.74	.2460
120	5.966	135.38	.2766	2.943	135.00	.2628	1.935	134.61	.2546	1.430	134.20	.2486
130	6.072	136.82	.2791	2.998	136.45	.2653	1.972	136.06	.2571	1.459	135.67	.2511
140	6.178	138.28	.2815	3.052	137.91	.2678	2.009	137.53	.2596	1.487	137.15	.2536
150	6.285	139.75	0.2840	3.106	139.39	0.2702	2.046	139.02	0.2620	1.515	138.65	0.2561
160	6.391	141.24	.2864	3.160	140.90	.2727	2.083	140.53	.2645	1.543	140.17	.2585
170	6.497	142.75	.2888	3.214	142.41	.2751	2.119	142.06	.2669	1.572	141.70	.2610
180	6.603	144.27	.2912	3.268	143.93	.2775	2.156	143.59	.2693	1.600	143.24	.2634
190	6.709	145.79	.2936	3.322	146.46	.2799	2.193	145.13	.2717	1.628	144.79	.2658
200	6.815	147.33	0.2959	3.376	147.01	0.2822	2.229	146.68	0.2741	1.656	146.35	0.2682
210	6.921	148.89	.2983	3.430	148.58	.2846	2.266	148.25	.2764	1.683	147.93	.2706
220	7.027	150.46	.3006	3.484	150.16	.2869	2.302	149.84	.2788	1.711	149.52	.2729
230	7.133	152.05	.3029	3.537	151.75	.2892	2.338	151.44	.2811	1.739	151.13	.2753
240	7.329	153.65	.3052	3.591	153.35	.2915	2.375	153.05	.2834	1.767	152.75	.2776
250	7.344	155.26	0.3075	3.645	154.97	0.2938	2.411	154.68	0.2857	1.794	154.38	0.2799
260	7.450	156.88	.3098	3.698	156.60	.2961	2.447	156.31	.2880	1.822	156.01	.2822
270	7.555	158.52	.3120	3.752	158.24	.2984	2.483	157.95	.2903	1.849	157.66	.2845
280	7.661	160.17	.3143	3.805	159.90	.3007	2.520	159.61	.2926	1.877	159.33	.2868
290	7.767	161.84	.3165	3.859	161.57	.3029	2.556	161.30	.2948	1.904	161.02	.2890

Temp F	Pressure 50 psia Gage pressure 35.3 psig (Sat temp 113.3 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 124.7 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 134.7 F)			Pressure 80 psia Gage pressure 65.3 psig (Sat temp 143.8 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(1.111)*	*(132.79)*	*(0.2421)*	*(0.9333)*	*(134.03)*	*(0.2410)*	*(0.8046)*	*(135.10)*	*(0.2401)*	*(0.7069)*	*(136.06)*	*(0.2393)*
150	1.197	138.26	0.2513	0.9836	137.86	0.2474	0.8312	137.45	0.2440	0.7167	137.01	0.2409
160	1.220	139.79	.2538	1.003	139.49	.2499	.8486	139.01	.2465	.7321	138.58	.2435
170	1.243	141.33	.2563	1.023	140.95	.2524	.8658	140.58	.2490	.7476	140.16	.2460
180	1.265	142.88	.2587	1.042	142.51	.2548	.8828	142.13	.2515	.7629	141.74	.2485
190	1.288	144.43	.2612	1.062	144.07	.2573	.8998	143.71	.2539	.7781	143.33	.2509
200	1.311	146.00	0.2636	1.081	145.65	0.2597	0.9168	145.30	0.2564	0.7933	144.93	0.2534
210	1.334	147.59	.2659	1.100	147.25	.2621	.9336	146.90	.2588	.8083	146.54	.2558
220	1.356	149.19	.2683	1.120	148.86	.2645	.9504	148.52	.2612	.8233	148.17	.2582
230	1.379	150.80	.2707	1.139	150.48	.2668	.9671	150.15	.2635	.8382	149.81	.2606
240	1.401	152.42	.2730	1.158	152.11	.2692	.9837	151.79	.2659	.8530	151.46	.2630
250	1.424	154.06	0.2753	1.177	153.76	0.2715	1.000	153.44	0.2682	0.8678	153.12	0.2654
260	1.446	155.71	.2776	1.196	155.41	.2738	1.017	155.10	.2706	.8825	154.78	.2677
270	1.469	157.37	.2799	1.215	157.08	.2761	1.033	156.77	.2729	.8971	156.46	.2700
280	1.491	159.05	.2822	1.234	158.76	.2784	1.050	158.45	.2752	.9117	158.16	.2723
290	1.513	160.74	.2845	1.252	160.45	.2807	1.066	160.16	.2775	.9262	159.87	.2746
300	1.535	162.44	0.2867	1.271	162.16	0.2830	1.082	161.88	0.2797	0.9406	161.59	0.2769
310	1.558	164.16	.2890	1.290	163.88	.2852	1.099	163.61	.2820	.9550	163.32	.2792
320	1.580	165.89	.2912	1.309	165.62	.2875	1.115	165.35	.2842	.9694	165.07	.2814
330	1.602	167.63	.2934	1.327	167.36	.2897	1.131	167.10	.2865	.9838	166.83	.2837
340	1.624	169.38	.2956	1.346	169.12	.2919	1.147	168.86	.2887	.9981	168.60	.2859

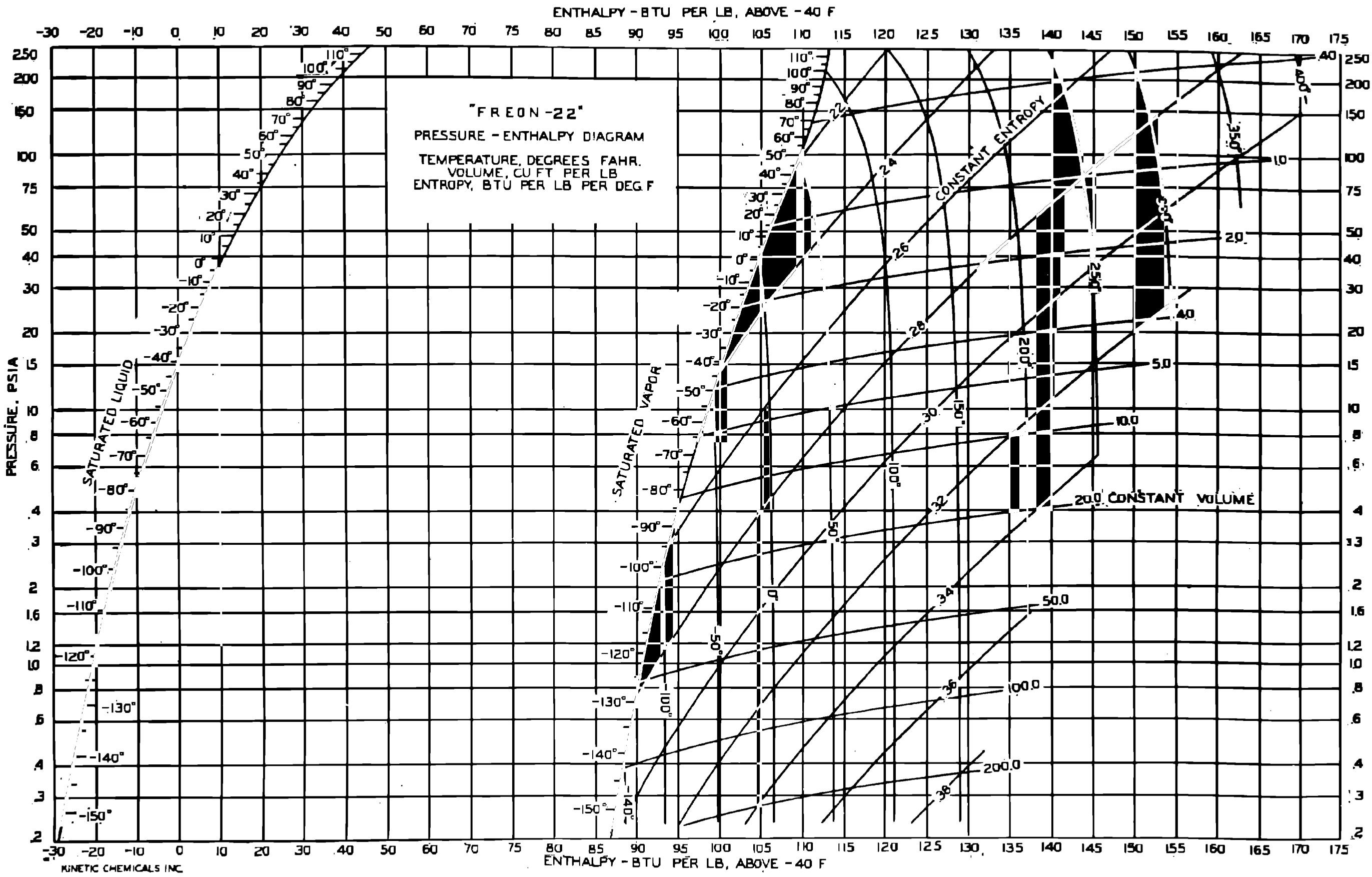

"FREON-22"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR.
VOLUME, CU FT PER LB
ENTROPY, BTU PER LB PER DEG F
ENTHALPY - BTU PER LB, ABOVE -40 F
PRESSURE, PSIA
SATURATED LIQUID
SATURATED VAPOR
CONSTANT ENTROPY
CONSTANT VOLUME
-30 -20 -10 0 10 20 30 40 50 60 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140 145 150 155 160 165 170 175
250 200 150 100 75 50 40 30 20 15 10 8 6 4 3 2 1.6 1.2 1.0 .8 .6 .4 .3 .2
110° 100° 90° 80° 70° 60° 50° 40° 30° 20° 10° 0° -10° -20° -30° -40° -50° -60° -70° -80° -90° -100° -110° -120° -130° -140° -150°
400° 350° 300° 250° 200° 150° 100° 50° 0° -50° -100° -140°
.22 .24 .26 .28 .30 .32 .34 .36 .38
.40 1.0 2.0 4.0 5.0 10.0 20.0 50.0 100.0 200.0
KINETIC CHEMICALS INC.

Chart 10

Table 20. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−155	0.19901	29.51*	97.67	188.1	−29.07	86.78	−0.0808	0.2996
−150	0.2605	29.39*	97.33	146.1	−27.79	87.36	−0.0767	0.2952
−145	0.3375	29.23*	96.99	114.5	−26.52	87.94	− .0727	.2912
−140	0.4332	29.04*	96.63	90.61	−25.25	88.53	−0.0687	0.2874
−135	0.5511	28.80*	96.27	72.33	−23.99	89.11	− .0647	.2837
−130	0.6949	28.51*	95.91	58.21	−22.73	89.70	−0.0609	0.2803
−125	0.8692	28.15*	95.53	47.23	−21.47	90.29	− .0571	.2770
−120	1.079	27.72*	95.15	38.60	−20.22	90.88	−0.0534	0.2738
−115	1.329	27.21*	94.76	31.77	−18.98	91.47	− .0497	.2708
−110	1.626	26.61*	94.37	26.33	−17.73	92.07	−0.0461	0.2680
−105	1.976	25.90*	93.97	21.96	−16.48	92.67	− .0425	.2653
−100	2.386	25.06*	93.56	18.43	−15.23	93.27	−0.0390	0.2627
− 95	2.865	24.09*	93.14	15.54	−13.98	93.87	− .0356	.2602
− 90	3.417	22.96*	92.72	13.20	−12.73	94.47	−0.0322	0.2579
− 85	4.055	21.67*	92.29	11.26	−11.47	95.08	− .0288	.2556
− 80	4.787	20.18*	91.85	9.650	−10.22	95.68	−0.0255	0.2535
− 78	5.100	19.55*	91.67	9.086	− 9.72	95.92	− .0242	.2526
− 76	5.430	18.87*	91.49	8.561	− 9.21	96.16	− .0229	.2518
− 74	5.79	18.14*	91.31	8.072	− 8.70	96.40	− .0216	.2510
− 72	6.17	17.37*	91.13	7.616	− 8.20	96.64	− .0203	.2502
− 70	6.57	16.55*	90.95	7.192	− 7.69	96.88	−0.0253	0.2494
− 68	6.99	15.70*	90.77	6.795	− 7.19	97.12	− .0177	.2487
− 66	7.40	14.86*	90.58	6.426	− 6.68	97.36	− .0164	.2479
− 64	7.86	13.93*	90.39	6.079	− 6.17	97.60	− .0151	.2472
− 62	8.35	12.93*	90.21	5.755	− 5.67	97.84	− .0138	.2465
− 60	8.86	11.89*	90.03	5.452	− 5.16	98.08	−0.0126	0.2458
− 58	9.39	10.81*	89.84	5.166	− 4.65	98.32	− .0113	.2451
− 56	9.94	9.69*	89.65	4.900	− 4.13	98.56	− .0100	.2444
− 54	10.51	8.53*	89.46	4.650	− 3.61	98.80	− .0087	.2438
− 52	11.11	7.31*	89.27	4.415	− 3.09	99.04	− .0075	.2431
− 50	11.74	6.03*	89.08	4.192	− 2.58	99.28	−0.0062	0.2425
− 48	12.40	4.68*	88.88	3.986	− 2.06	99.52	− .0050	.2418
− 46	13.09	3.28*	88.68	3.793	− 1.54	99.76	− .0037	.2412
− 44	13.80	1.83*	88.49	3.611	− 1.02	100.00	− .0025	.2406
− 42	14.54	0.326*	88.30	3.440	− 0.51	100.23	− .0012	.2400
− 40	15.31	0.610	88.10	3.279	0.00	100.46	0.0000	0.2394
− 38	16.12	1.42	87.90	3.126	0.53	100.70	.0013	.2389
− 36	16.97	2.27	87.70	2.981	1.05	100.93	.0025	.2383
− 34	17.85	3.15	87.50	2.844	1.58	101.17	.0037	.2377
− 32	18.77	4.07	87.29	2.713	2.10	101.40	.0050	.2372
− 30	19.72	5.02	87.09	2.590	2.62	101.63	0.0062	0.2367
− 28	20.71	6.01	86.89	2.474	3.15	101.86	.0074	.2361
− 26	21.73	7.03	86.69	2.365	3.69	102.10	.0086	.2356
− 24	22.79	8.09	86.48	2.262	4.22	102.33	.0099	.2351
− 22	23.88	9.18	86.27	2.165	4.75	102.56	.0111	.2346
− 20	25.01	10.31	86.06	2.074	5.28	102.79	0.0123	0.2341
− 18	26.18	11.48	85.85	1.987	5.82	103.02	.0135	.2336
− 16	27.39	12.69	85.64	1.905	6.40	103.25	.0147	.2331
− 14	28.64	13.94	85.43	1.827	6.90	103.48	.0159	.2326
− 12	29.94	15.24	85.21	1.752	7.43	103.70	.0170	.2321
− 10	31.29	16.59	84.99	1.681	7.96	103.92	0.0182	0.2316
− 8	32.69	17.99	84.78	1.613	8.49	104.14	.0194	.2312
− 6	34.14	19.44	84.56	1.549	9.02	104.36	.0205	.2307
− 4	35.64	20.94	84.34	1.488	9.55	104.58	.0217	.2302
− 2	37.19	22.49	84.12	1.429	10.09	104.80	.0228	.2298
0	38.79	24.09	83.90	1.373	10.63	105.02	0.0240	0.2293
2	40.43	25.73	83.68	1.320	11.17	105.24	0.0251	0.2289
4	42.14	27.44	83.45	1.270	11.70	105.45	.0262	.2285
5	43.02	28.33	83.34	1.246	11.97	105.56	.0268	.2283
6	43.91	29.21	83.23	1.221	12.23	105.66	.0274	.2280
8	45.74	31.04	83.01	1.175	12.76	105.87	.0285	.2276
10	47.63	32.93	82.78	1.130	13.29	106.08	0.0296	0.2272
12	49.58	34.88	82.55	1.088	13.82	106.29	.0307	.2268
14	51.59	36.89	82.32	1.048	14.36	106.50	.0319	.2264
16	53.66	38.96	82.09	1.009	14.90	106.71	.0330	.2260
18	55.79	41.09	81.86	0.9721	15.44	106.92	.0341	.2257

* Inches mercury below one atmosphere.

Table 20. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Liquid and Saturated Vapor (Concluded)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
20	57.98	43.28	81.63	0.9369	15.98	107.13	0.0352	0.2253
22	60.23	45.53	81.39	.9032	16.52	107.33	.0364	.2249
24	62.55	47.85	81.16	.8707	17.06	107.53	.0375	.2246
26	64.94	50.24	80.92	.8398	17.61	107.73	.0379	.2242
28	67.40	52.70	80.69	.8100	18.17	107.93	.0398	.2239
30	69.93	55.23	80.45	0.7816	18.74	108.13	0.0409	0.2235
32	72.53	57.83	80.21	.7543	19.32	108.33	.0421	.2232
34	75.21	60.51	79.97	.7283	19.90	108.52	.0433	.2228
36	77.97	63.27	79.73	.7032	20.49	108.71	.0445	.2225
38	80.81	66.11	79.49	.6791	21.09	108.90	.0457	.2222
40	83.72	69.02	79.25	0.6559	21.70	109.09	0.0469	0.2218
42	86.69	71.99	79.00	.6339	22.29	109.27	.0481	.2215
44	89.74	75.04	78.76	.6126	22.90	109.45	.0493	.2211
46	92.88	78.18	78.51	.5922	23.50	109.63	.0505	.2208
48	96.10	81.40	78.26	.5726	24.11	109.80	.0516	.2205
50	99.40	84.70	78.02	0.5537	24.73	109.98	0.0528	0.2201
52	102.8	88.10	77.77	.5355	25.34	110.14	.0540	.2198
54	106.2	91.5	77.51	.5184	25.95	110.30	.0552	.2194
56	109.8	95.1	77.26	.5014	26.58	110.47	.0564	.2191
58	113.5	98.8	77.01	.4849	27.22	110.63	.0576	.2188
60	117.2	102.5	76.75	0.4695	27.83	110.78	0.0588	0.2185
62	121.0	106.3	76.50	.4546	28.46	110.93	.0600	.2181
64	124.9	110.2	76.24	.4403	29.09	111.08	.0612	.2178
66	128.9	114.2	75.98	.4264	29.72	111.22	.0624	.2175
68	133.0	118.3	75.72	.4129	30.35	111.35	.0636	.2172
70	137.2	122.5	75.46	0.4000	30.99	111.49	0.0648	0.2168
72	141.5	126.8	75.20	.3875	31.65	111.63	.0661	.2165
74	145.9	131.2	74.94	.3754	32.29	111.75	.0673	.2162
76	150.4	135.7	74.68	.3638	32.94	111.88	.0684	.2158
78	155.0	140.3	74.41	.3526	33.61	112.01	.0696	.2155
80	159.7	145.0	74.15	0.3417	34.27	112.13	0.0708	0.2151
82	164.5	149.8	73.89	.3313	34.92	112.24	.0720	.2148
84	169.4	154.7	73.63	.3212	35.60	112.36	.0732	.2144
86	174.5	159.8	73.36	.3113	36.28	112.47	.0744	.2140
88	179.6	164.9	73.09	.3019	36.94	112.57	.0756	.2137
90	184.8	170.1	72.81	0.2928	37.61	112.67	0.0768	0.2133
92	190.1	175.4	72.53	.2841	38.28	112.76	.0780	.2130
94	195.6	180.9	72.24	.2755	38.97	112.85	.0792	.2126
96	201.2	186.5	71.95	.2672	39.65	112.93	.0803	.2122
98	206.8	192.1	71.65	.2594	40.32	113.00	.0815	.2119
100	212.6	197.9	71.35	0.2517	40.98	113.06	0.0827	0.2115
102	218.5	203.8	71.05	.2443	41.65	113.12	.0839	.2111
104	224.6	209.9	70.74	.2370	42.32	113.16	.0851	.2107
106	230.7	216.0	70.42	.2301	42.98	113.20	.0862	.2104
108	237.0	222.3	70.11	.2233	43.66	113.24	.0874	.2100
110	243.4	228.7	69.78	0.2167	44.35	113.29	0.0886	0.2096
112	249.9	235.2	69.45	.2104	45.04	113.34	.0898	.2093
114	256.6	241.9	69.12	.2043	45.74	113.38	.0909	.2089
116	263.4	248.7	68.78	.1983	46.44	113.42	.0921	.2085
118	270.3	255.6	68.44	.1926	47.14	113.46	.0933	.2081
120	277.3	262.6	68.10	0.1871	47.85	113.52	0.0945	0.2078
122	384.4	269.7	67.75	.1825	48.6	113.57		
124	391.6	276.9	67.40	.1772	49.4	113.61		
126	398.8	284.1	67.05	.1724	50.2	113.65		
128	306.1	291.4	66.70	.1675	50.8	113.69		
130	313.5	298.8	66.35	0.1629	51.5	113.71		
132	321.0	306.3	66.00	.1585	52.3	113.74		
134	328.7	314.0	65.65	.1538	53.1	113.77		
136	336.6	321.9	65.25	.1492	53.8	113.79		
138	344.6	329.9	64.85	.1449	54.6	113.80		
140	352.7	338.0	64.45	0.1408	55.3	113.81		
142	361.0	346.3	64.05	.1368	56.1	113.80		
144	369.7	355.0	63.65	.1330	56.9	113.79		
146	379.0	364.3	63.25	.1292	57.7	113.78		
148	388.8	374.1	62.85	.1253	58.4	113.76		
150	399.0	384.3	62.45	0.1216	59.2	113.74		
152	407.0	392.3	62.02	.1179	60.0	113.71		
154	416.0	401.3	61.58	.1141	60.8	113.67		
156	426.0	411.3	61.13	.1105	61.6	113.62		
158	436.5	421.8	60.67	.1070	62.5	113.56		
160	448.0	433.3	60.20	0.1035	63.5	113.50		

Table 21. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Superheated Vapor

Temp F	Pressure .25 psia Gage pressure 29.41 in. vac (Sat temp −150.9 F)			Pressure .30 psia Gage pressure 29.31 in. vac (Sat temp −147.4 F)			Pressure .35 psia Gage pressure 29.21 in. vac (Sat temp −144.3 F)			Pressure .40 psia Gage pressure 29.11 in. vac (Sat temp −141.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(151.8)*	*(87.26)*	*(0.2960)*	*(127.9)*	*(87.66)*	*(0.2932)*	*(110.7)*	*(88.02)*	*(0.2907)*	*(97.62)*	*(88.33)*	*(0.2887)*
−150	152.19	87.36	.2962									
−140	157.12	88.53	.2999	130.91	88.53	.2958	112.19	88.53	.2923	98.15	88.53	.2892
−130	162.04	89.72	.3036	135.01	89.72	.2995	115.71	89.72	.2959	101.23	89.71	.2929
−120	166.96	90.93	.3072	139.12	90.92	.3031	119.23	90.92	.2995	104.31	90.92	.2965
−110	171.89	92.14	.3107	143.22	92.14	.3066	122.75	92.14	.3031	107.39	92.14	.3000
−100	176.81	93.38	.3142	147.33	93.37	.3101	126.26	93.37	.3065	110.47	93.37	.3035
− 90	181.74	94.62	.3176	151.43	94.62	.3135	129.78	94.62	.3100	113.55	94.62	.3069
− 80	186.66	95.89	.3210	155.53	95.88	.3168	133.30	95.88	.3133	116.62	95.88	.3103
− 70	191.59	97.16	.3243	159.64	97.16	.3201	136.82	97.16	.3166	119.70	97.16	.3136
− 60	196.51	98.46	.3276	163.74	98.45	.3234	140.33	98.45	.3199	122.78	98.45	.3168
− 50	201.43	99.76	.3308	167.84	99.76	.3266	143.85	99.76	.3231	125.86	99.75	.3201
− 40	206.35	101.08	.3340	171.95	101.08	.3298	147.37	101.08	.3263	128.94	101.08	.3233
− 30	211.28	102.42	.3371	176.05	102.42	.3330	150.89	102.41	.3295	132.01	102.41	.3264
− 20	216.20	103.77	.3403	180.15	103.77	.3361	154.40	103.77	.3326	135.09	103.76	.3295
− 10	221.12	105.14	.3433	184.25	105.13	.3392	157.92	105.13	.3357	138.17	105.13	.3326
0	226.04	106.52	.3464	188.36	106.51	.3422	161.44	106.51	.3387	141.25	106.51	.3357
10	230.97	107.91	.3494	192.46	107.91	.3452	164.96	107.91	.3417	144.33	107.91	.3387
20	235.89	109.32	.3524	196.56	109.32	.3482	168.47	109.32	.3447	147.40	109.32	.3417
30	240.81	110.75	.3553	200.66	110.75	.3512	171.99	110.74	.3476	150.48	110.74	.3446
40	245.73	112.19	.3582	204.76	112.18	.3541	175.50	112.18	.3506	153.56	112.18	.3475
50	250.66	113.64	.3611	208.87	113.64	.3569	179.02	113.64	.3534	156.63	113.64	.3504
60	255.58	115.11	.3639	212.97	115.11	.3598	182.54	115.11	.3563	159.71	115.10	.3532
70	260.50	116.59	.3668	217.07	116.59	.3626	186.05	116.59	.3591	162.79	116.59	.3561
80	265.42	118.09	.3696	221.17	118.09	.3654	189.57	118.09	.3619	165.87	118.09	.3589
90	270.34	119.61	.3724	225.28	119.60	.3682	193.09	119.60	.3647	168.94	119.60	.3617
100	275.27	121.13	.3751	229.38	121.13	.3710	196.60	121.13	.3675	172.02	121.13	.3644
110	280.19	122.68	.3779	233.48	122.68	.3737	200.12	122.68	.3702	175.10	122.67	.3672
120	285.11	124.24	.3806	237.58	124.23	.3764	203.63	124.23	.3729	178.17	124.23	.3699
130	290.03	125.81	.3833	241.68	125.81	.3791	207.15	125.81	.3756	181.25	125.80	.3726
140	294.95	127.40	.3859	245.78	127.40	.3818	210.66	127.39	.3783	184.33	127.39	.3752
150	299.87	129.00	.3886	249.89	129.00	.3844	214.18	129.00	.3809	187.40	129.00	.3779
160				253.99	130.62	.3871	217.70	130.61	.3835	190.48	130.61	.3805

Temp F	Pressure .45 psia Gage pressure 29.00 in. vac (Sat temp −139.3 F)			Pressure .50 psia Gage pressure 28.90 in. vac (Sat temp −137.2 F)			Pressure .55 psia Gage pressure 28.80 in. vac (Sat temp −135.0 F)			Pressure .60 psia Gage pressure 28.70 in. vac (Sat temp −133.3 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(87.42)*	*(88.61)*	*(0.2868)*	*(79.18)*	*(88.86)*	*(0.2853)*	*(72.47)*	*(89.10)*	*(0.2838)*	*(66.77)*	*(89.31)*	*(0.2825)*
−130	89.97	89.71	.2902	80.96	89.71	.2878	73.59	89.71	.2856	67.44	89.70	.2837
−120	92.70	90.92	.2938	83.42	90.91	.2914	75.83	90.91	.2892	69.50	90.91	.2873
−110	95.44	92.13	.2974	85.89	92.13	.2949	78.07	92.13	.2928	71.55	92.13	.2908
−100	98.18	93.37	.3008	88.35	93.36	.2984	80.31	93.36	.2962	73.61	93.36	.2943
− 90	100.92	94.61	.3042	90.81	94.61	.3018	82.55	94.91	.2997	75.66	94.61	.2977
− 80	103.65	95.88	.3076	93.28	95.87	.3052	84.79	95.87	.3031	77.71	95.87	.3010
− 70	106.39	97.15	.3109	95.74	97.15	.3085	87.03	97.15	.3063	79.77	97.15	.3043
− 60	109.13	98.45	.3142	98.21	98.44	.3118	89.27	98.44	.3096	81.32	98.44	.3076
− 50	111.86	99.75	.3174	100.67	99.75	.3150	91.51	99.75	.3128	83.88	99.75	.3108
− 40	114.60	101.07	.3206	103.13	101.07	.3182	93.75	101.07	.3160	85.93	101.07	.3140
− 30	117.34	102.41	.3237	105.60	102.41	.3213	95.99	102.41	.3192	87.98	102.40	.3172
− 20	120.07	103.76	.3269	108.06	103.76	.3245	98.23	103.76	.3223	90.03	103.76	.3203
− 10	122.81	105.13	.3299	110.52	105.13	.3275	100.46	105.12	.3254	92.09	105.12	.3234
0	125.54	106.51	.3330	112.98	106.51	.3306	102.70	106.50	.3284	94.14	106.50	.3264
10	128.28	107.90	.3360	115.45	107.90	.3336	104.94	107.90	.3314	96.19	107.90	.3294
20	131.02	109.31	.3390	117.91	109.31	.3366	107.18	109.31	.3344	98.24	109.31	.3324
30	133.75	110.74	.3419	120.37	110.74	.3395	109.42	110.74	.3373	100.30	110.73	.3354
40	136.49	112.18	.3448	122.83	112.18	.3424	111.66	112.18	.3403	102.35	112.17	.3383
50	139.22	113.63	.3477	125.29	113.63	.3453	113.90	113.63	.3431	104.40	113.63	.3412
60	141.96	115.10	.3506	127.76	115.10	.3482	116.14	115.10	.3460	106.45	115.10	.3440
70	144.69	116.59	.3534	130.22	116.59	.3510	118.37	116.58	.3488	108.50	116.58	.3468
80	147.43	118.09	.3562	132.68	118.09	.3538	120.61	118.08	.3516	110.55	118.08	.3496
90	150.16	119.60	.3590	135.14	119.60	.3566	122.85	119.60	.3544	112.61	119.60	.3524
100	152.90	121.13	.3617	137.60	121.13	.3593	125.09	121.13	.3572	114.66	121.12	.3552
110	155.63	122.67	.3645	140.07	122.67	.3621	127.33	122.67	.3599	116.71	122.67	.3579
120	158.37	124.23	.3672	142.53	124.23	.3648	129.56	124.23	.3626	118.76	124.23	.3606
130	161.10	125.80	.3699	144.99	125.80	.3675	131.80	125.80	.3653	120.81	125.80	.3633
140	163.84	127.39	.3725	147.45	127.39	.3701	134.04	127.39	.3680	122.87	127.39	.3660
150	166.57	128.99	.3752	149.91	128.99	.3728	136.28	128.99	.3706	124.92	128.99	.3686
160	169.31	130.61	.3778	152.37	130.61	.3754	138.52	130.61	.3732	126.97	130.61	.3713
170	172.04	132.24	.3804	154.83	132.24	.3780	140.75	132.24	.3759	129.02	132.24	.3739

Table 21. F-22 (CHClF$_2$) Monochlorodifluoromethane—Properties of Superheated Vapor (Continued)

Temp F	Pressure .65 psia Gage pressure 28.60 in. vac (Sat temp −131.6 F)			Pressure .70 psia Gage pressure 28.50 in. vac (Sat temp −129.9 F)			Pressure .75 psia Gage pressure 28.39 in. vac (Sat temp −128.4 F)			Pressure .80 psia Gage pressure 28.29 in. vac (Sat temp −127.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(61.94)*	*(89.52)*	*(0.2813)*	*(57.81)*	*(89.72)*	*(0.2802)*	*(54.19)*	*(89.89)*	*(0.2792)*	*(51.01)*	*(90.06)*	*(0.2783)*
−130	62.25	89.70	.2818									
−120	64.14	90.90	.2854	59.55	90.90	.2837	55.58	90.90	.2822	52.09	90.90	.2807
−110	66.04	92.12	.2890	61.31	92.12	.2873	57.22	92.12	.2857	53.64	92.11	.2842
−100	67.94	93.36	.2924	63.08	93.35	.2907	58.86	93.35	.2892	55.18	93.35	.2877
− 90	69.83	94.60	.2958	64.84	94.60	.2942	60.51	94.60	.2926	56.72	94.60	.2911
− 80	71.73	95.87	.2992	66.60	95.87	.2975	62.15	95.86	.2959	58.26	95.86	.2945
− 70	73.62	97.14	.3025	68.36	97.14	.3005	63.79	97.14	.2992	59.80	97.14	.2978
− 60	75.52	98.44	.3058	70.12	98.44	.3041	65.44	98.43	.3025	61.34	98.43	.3010
− 50	77.42	99.74	.3090	71.88	99.74	.3073	67.08	99.74	.3057	62.78	99.74	.3043
− 40	79.31	101.07	.3122	73.64	101.06	.3105	68.73	101.06	.3089	64.42	101.06	.3075
− 30	81.21	102.40	.3154	75.40	102.40	.3137	70.37	102.40	.3121	65.96	102.40	.3106
− 20	83.10	103.75	.3185	77.16	103.75	.3168	72.01	103.75	.3152	67.50	103.75	.3137
− 10	85.00	105.12	.3216	78.92	105.12	.3199	73.65	105.12	.3183	69.04	105.11	.3168
0	86.89	106.50	.3246	80.68	106.50	.3229	75.30	106.50	.3213	70.58	106.50	.3199
10	88.79	107.90	.3276	82.44	107.89	.3259	76.94	107.89	.3244	72.12	107.89	.3229
20	90.68	109.31	.3306	84.20	109.31	.3289	78.58	109.30	.3273	73.66	109.30	.3259
30	92.58	110.73	.3335	85.96	110.73	.3318	80.22	110.73	.3303	75.20	110.73	.3288
40	94.47	112.17	.3364	87.72	112.17	.3348	81.86	112.17	.3332	76.74	112.17	.3317
50	96.36	113.63	.3393	89.48	113.63	.3376	83.51	113.62	.3361	78.28	113.62	.3346
60	98.26	115.10	.3422	91.23	115.09	.3405	85.15	115.09	.3389	79.82	115.09	.3374
70	100.15	116.58	.3450	92.99	116.58	.3433	86.79	116.58	.3417	81.36	116.58	.3403
80	102.05	118.08	.3478	94.75	118.08	.3461	88.43	118.08	.3446	82.90	118.08	.3431
90	103.94	119.59	.3506	96.51	119.59	.3489	90.07	119.59	.3473	84.44	119.59	.3459
100	105.84	121.12	.3534	98.27	121.12	.3517	91.72	121.12	.3501	85.98	121.12	.3486
110	107.73	122.67	.3561	100.03	122.67	.3544	93.36	122.66	.3528	87.52	122.66	.3514
120	109.62	124.22	.3588	101.79	124.22	.3571	95.00	124.22	.3555	89.06	124.22	.3541
130	111.52	125.80	.3615	103.55	125.80	.3598	96.64	125.80	.3582	90.60	125.79	.3568
140	113.41	127.39	.3642	105.31	127.38	.3625	98.28	127.38	.3609	92.14	127.38	.3594
150	115.30	128.99	.3668	107.06	128.99	.3651	99.92	128.99	.3636	93.68	128.98	.3612
160	117.20	130.61	.3694	108.82	130.60	.3678	101.56	130.60	.3662	95.21	130.60	.3647
170	119.09	132.24	.3721	110.58	132.24	.3704	103.21	132.24	.3688	96.75	132.23	.3673
180				112.34	133.88	.3730	104.85	133.88	.3714	98.29	133.88	.3699

Temp F	Pressure .90 psia Gage pressure 28.09 in. vac (Sat temp −124.3 F)			Pressure 1.00 psia Gage pressure 27.88 in. vac (Sat temp −121.9 F)			Pressure 1.25 psia Gage pressure 27.38 in. vac (Sat temp −116.6 F)			Pressure 1.50 psia Gage pressure 26.87 in. vac (Sat temp −112.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(45.70)*	*(90.38)*	*(0.2765)*	*(41.42)*	*(90.66)*	*(0.2750)*	*(33.63)*	*(91.28)*	*(0.2718)*	*(28.38)*	*(91.82)*	*(0.2692)*
−120	46.29	90.89	.2780	41.65	90.88	.2756						
−110	47.66	92.11	.2815	42.89	92.10	.2791	34.29	92.09	.2740	28.55	92.08	.2699
−100	49.03	93.34	.2850	44.12	93.34	.2826	35.27	93.33	.2775	29.38	93.31	.2733
− 90	50.41	94.59	.2884	45.35	94.59	.2860	36.26	94.58	.2809	30.20	94.56	.2767
− 80	51.78	95.86	.2918	46.59	95.85	.2894	37.25	95.84	.2843	31.02	95.83	.2801
− 70	53.15	97.13	.2951	47.82	97.13	.2927	38.24	97.12	.2876	31.85	97.11	.2834
− 60	54.52	98.43	.2984	49.05	98.42	.2960	39.22	98.41	.2909	32.67	98.40	.2867
− 50	55.89	99.73	.3016	50.29	99.73	.2992	40.21	99.72	.2941	33.49	99.71	.2899
− 40	57.26	101.06	.3048	51.52	101.05	.3024	41.20	101.04	.2973	34.32	101.03	.2931
− 30	58.63	102.39	.3079	52.75	102.39	.3055	42.19	102.38	.3004	35.14	102.37	.2963
− 20	59.99	103.74	.3111	53.99	103.74	.3087	43.17	103.73	.3036	35.96	103.72	.2994
− 10	61.36	105.11	.3141	55.22	105.11	.3117	44.16	105.10	.3066	36.79	105.09	.3025
0	62.73	106.49	.3172	56.45	106.49	.3148	45.15	106.48	.3097	37.61	106.47	.3055
10	64.10	107.89	.3202	57.68	107.88	.3178	46.13	107.87	.3127	38.43	107.86	.3085
20	65.47	109.30	.3232	58.92	109.29	.3208	47.12	109.29	.3157	39.25	109.28	.3115
30	66.84	110.72	.3261	60.15	110.72	.3237	48.10	110.71	.3186	40.08	110.70	.3145
40	68.21	112.16	.3290	61.38	112.16	.3266	49.09	112.15	.3215	40.90	112.14	.3174
50	69.58	113.62	.3319	62.61	113.62	.3295	50.08	113.61	.3244	41.72	113.60	.3203
60	70.95	115.09	.3348	63.84	115.08	.3324	51.06	115.07	.3273	42.54	115.06	.3231
70	72.31	116.57	.3376	65.08	116.57	.3352	52.05	116.56	.3301	43.36	116.55	.3259
80	73.68	118.07	.3404	66.31	118.07	.3380	53.03	118.06	.3329	44.18	118.05	.3287
90	75.05	119.59	.3432	67.54	119.58	.3408	54.02	119.58	.3357	45.01	119.57	.3315
100	76.42	121.12	.3459	68.77	121.11	.3435	55.01	121.10	.3384	45.83	121.10	.3343
110	77.79	122.66	.3487	70.00	122.66	.3463	55.99	122.65	.3412	46.65	122.64	.3370
120	79.16	124.22	.3514	71.24	124.21	.3490	56.98	124.21	.3439	47.47	124.20	.3397
130	80.53	125.79	.3541	72.47	125.79	.3517	57.96	125.78	.3466	48.29	125.77	.3424
140	81.89	127.38	.3567	73.70	127.38	.3543	58.95	127.37	.3493	49.12	127.36	.3451
150	83.26	128.99	.3594	74.93	128.99	.3570	59.93	128.97	.3519	49.94	128.97	.3478
160	84.63	130.60	.3620	76.10	130.60	.3596	60.92	130.59	.3545	50.76	130.58	.3504
170	86.00	132.23	.3646	77.39	132.23	.3622	61.90	132.22	.3571	51.58	132.22	.3530
180	87.37	133.88	.3762	78.62	133.88	.3648	62.89	133.87	.3597	52.40	133.86	.3556
190							63.88	135.53	.3623	53.22	135.53	.3582

Table 21. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Superheated Vapor (Continued)

Temp F	Pressure 1.75 psia Gage pressure 26.36 in. vac (Sat temp −108.2 F)			Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −104.7 F)			Pressure 2.5 psia Gage pressure 24.83 in. vac (Sat temp −98.8 F)			Pressure 3.0 psia Gage pressure 23.81 in. vac (Sat temp −93.8 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(24.58)*	*(92.28)*	*(0.2670)*	*(21.71)*	*(92.72)*	*(0.2651)*	*(17.64)*	*(93.41)*	*(0.2621)*	*(14.89)*	*(94.02)*	*(0.2597)*
−100	25.16	93.30	.2698	22.00	93.29	.2667						
−90	25.87	94.55	.2732	22.62	94.54	.2702	18.08	94.51	.2650	15.05	94.49	.2609
−80	26.58	95.82	.2766	23.24	95.81	.2735	18.57	95.78	.2684	15.46	95.76	.2642
−70	27.28	97.10	.2799	23.86	97.08	.2768	19.07	97.06	.2717	15.87	97.04	.2676
−60	27.99	98.39	.2832	24.48	98.38	.2801	19.56	98.36	.2750	16.29	98.34	.2708
−50	28.70	99.70	.2864	25.10	99.69	.2833	20.06	99.67	.2782	16.70	99.65	.2741
−40	29.40	101.02	.2896	25.72	101.01	.2865	20.55	100.99	.2814	17.11	100.97	.2773
−30	30.11	102.36	.2927	26.33	102.35	.2897	21.05	102.33	.2846	17.53	102.31	.2804
−20	30.81	103.71	.2959	26.95	103.70	.2928	21.54	103.68	.2877	17.94	103.66	.2835
−10	31.52	105.08	.2990	27.57	105.07	.2959	22.04	105.05	.2908	18.35	105.03	.2866
0	32.22	106.46	.3020	28.19	106.45	.2990	22.53	106.43	.2939	18.76	106.41	.2897
10	32.93	107.85	.3050	28.80	107.85	.3020	23.03	107.83	.2969	19.18	107.81	.2927
20	33.63	109.27	.3080	29.42	109.26	.3049	23.52	109.24	.2998	19.59	109.22	.2957
30	34.34	110.69	.3109	30.04	110.68	.3079	24.02	110.67	.3028	20.00	110.65	.2986
40	35.04	112.13	.3139	30.66	112.13	.3108	24.51	112.11	.3057	20.41	112.09	.3015
50	35.75	113.59	.3167	31.27	113.58	.3137	25.00	113.57	.3086	20.83	113.55	.3044
60	36.45	115.06	.3196	31.89	115.05	.3165	25.50	115.04	.3114	21.24	115.02	.3073
70	37.16	116.55	.3224	32.51	116.54	.3194	25.99	116.52	.3143	21.65	116.51	.3101
80	37.86	118.05	.3252	33.12	118.04	.3222	26.49	118.02	.3171	22.06	118.01	.3129
90	38.57	119.56	.3280	33.74	119.55	.3250	26.98	119.54	.3199	22.47	119.52	.3157
100	39.27	121.09	.3308	34.36	121.08	.3277	27.47	121.07	.3226	22.88	121.05	.3185
110	39.98	122.63	.3335	34.97	122.63	.3305	27.97	122.61	.3254	23.30	122.60	.3212
120	40.68	124.19	.3362	35.59	124.19	.3332	28.46	124.17	.3281	23.71	124.16	.3239
130	41.39	125.77	.3389	36.21	125.76	.3359	28.95	125.75	.3308	24.12	125.73	.3266
140	42.09	127.36	.3416	36.82	127.35	.3385	29.45	127.34	.3334	24.53	127.32	.3293
150	42.80	128.96	.3442	37.44	128.95	.3412	29.94	128.94	.3361	24.94	128.93	.3319
160	43.50	130.58	.3469	38.06	130.57	.3438	30.44	130.56	.3387	25.35	130.54	.3346
170	44.20	132.21	.3495	38.67	132.20	.3464	30.93	132.19	.3413	25.77	132.18	.3372
180	44.91	133.86	.3521	39.29	133.85	.3490	31.42	133.84	.3439	26.18	133.83	.3398
190	45.61	135.52	.3547	39.91	135.51	.3516	31.91	135.50	.3465	26.59	135.49	.3424
200	46.32	137.20	.3572	40.52	137.19	.3542	32.41	137.18	.3491	27.10	137.17	.3449
210							32.90	138.87	.3516	27.41	138.86	.3475

Temp F	Pressure 3.5 psia Gage pressure 22.80 in. vac (Sat temp −89.4 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp −85.4 F)			Pressure 4.5 psia Gage pressure 20.76 in. vac (Sat temp −82.0 F)			Pressure 5.0 psia Gage pressure 19.74 in. vac (Sat temp −78.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(12.90)*	*(94.55)*	*(0.2576)*	*(11.40)*	*(95.03)*	*(0.2558)*	*(10.22)*	*(95.46)*	*(0.2543)*	*(9.266)*	*(95.83)*	*(0.2529)*
80	13.14	95.74	.2607	11.57	95.72	.2576	10.27	95.69	.2549			
70	13.59	97.02	.2640	11.88	97.00	.2609	10.55	96.97	.2582	9.483	96.95	.2558
−60	13.95	98.31	.2673	12.19	98.29	.2642	10.83	98.27	.2615	9.733	98.25	.2591
−50	14.30	99.62	.2705	12.50	99.60	.2674	11.10	99.58	.2647	9.983	99.56	.2623
−40	14.66	100.95	.2737	12.81	100.93	.2706	11.38	100.91	.2679	10.232	100.89	.2655
−30	15.01	102.29	.2769	13.12	102.27	.2738	11.66	102.25	.2711	10.481	102.23	.2687
−20	15.37	103.64	.2800	13.43	103.62	.2769	11.93	103.60	.2742	10.729	103.58	.2718
−10	15.72	105.01	.2831	13.74	104.99	.2800	12.21	104.98	.2773	10.978	104.96	.2749
0	16.07	106.40	.2861	14.05	106.38	.2831	12.48	106.36	.2804	11.226	106.34	.2780
10	16.43	107.80	.2892	14.36	107.78	.2861	12.76	107.76	.2834	11.475	107.74	.2810
20	16.78	109.21	.2921	14.67	109.19	.2891	13.04	109.17	.2864	11.723	109.16	.2840
30	17.13	110.63	.2951	14.98	110.62	.2920	13.31	110.60	.2893	11.972	110.59	.2869
40	17.49	112.07	.2980	15.29	112.06	.2950	13.59	112.05	.2922	12.220	112.03	.2898
50	17.84	113.53	.3009	15.60	113.52	.2978	13.86	113.50	.2951	12.468	113.49	.2927
60	18.19	115.00	.3037	15.91	114.99	.3007	14.14	114.98	.2980	12.715	114.96	.2956
70	18.55	116.49	.3066	16.22	116.48	.3035	14.41	116.46	.3008	12.963	116.45	.2984
80	18.90	117.99	.3094	16.53	117.98	.3063	14.69	117.97	.3036	13.211	117.95	.3012
90	19.25	119.51	.3122	16.84	119.50	.3091	14.96	119.48	.3064	13.460	119.47	.3040
100	19.61	121.04	.3149	17.15	121.03	.3119	15.24	121.01	.3092	13.707	121.00	.3068
110	19.96	122.58	.3177	17.46	122.57	.3146	15.51	122.56	.3119	13.955	122.55	.3095
120	20.31	124.14	.3204	17.77	124.13	.3173	15.79	124.12	.3146	14.202	124.11	.3122
130	20.67	125.72	.3231	18.08	125.70	.3200	16.06	125.69	.3173	14.450	125.68	.3149
140	21.02	127.31	.3258	18.39	127.29	.3227	16.34	127.28	.3200	14.697	127.27	.3176
150	21.37	128.91	.3284	18.69	128.90	.3254	16.61	128.89	.3227	14.945	128.88	.3202
160	21.73	130.53	.3310	19.00	130.52	.3280	16.89	130.51	.3253	15.192	130.49	.3229
170	22.08	132.17	.3337	19.31	132.15	.3306	17.16	132.14	.3279	15.440	132.13	.3255
180	22.43	133.81	.3363	19.62	133.80	.3332	17.44	133.79	.3305	15.687	133.78	.3281
190	22.78	135.48	.3388	19.93	135.46	.3358	17.71	135.45	.3331	15.934	135.44	.3307
200	23.14	137.15	.3414	20.24	137.14	.3383	17.98	137.13	.3357	16.181	137.12	.3332
210	23.49	138.85	.3439	20.55	138.84	.3409	18.26	138.82	.3382	16.429	138.81	.3358
220	23.84	140.56	.3465	20.86	140.54	.3434	18.53	140.53	.3407	16.676	140.52	.3383
230										16.923	142.24	.3408

Table 21. F-22 ($CHClF_2$) Monochlorodifluoromethane Properties of Superheated Vapor (Continued)

Temp F	Pressure 6 psia Gage pressure 17.71 in. vac (Sat temp −73.0 F)			Pressure 7 psia Gage pressure 15.67 in. vac (Sat temp −68.0 F)			Pressure 8 psia Gage pressure 13.64 in. vac (Sat temp −63.5 F)			Pressure 9 psia Gage pressure 11.60 in. vac (Sat temp −59.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(7.823)*	*(96.62)*	*(0.2506)*	*(6.781)*	*(97.12)*	*(0.2487)*	*(5.991)*	*(97.66)*	*(0.2471)*	*(5.370)*	*(98.14)*	*(0.2456)*
−70	7.886	96.91	.2516									
−60	8.095	98.20	.2549	6.924	98.16	.2513	6.046	98.12	.2482			
−50	8.303	99.52	.2581	7.103	99.48	.2545	6.204	99.44	.2514	5.503	99.39	.2487
−40	8.511	100.85	.2613	7.282	100.80	.2577	6.361	100.76	.2546	5.643	100.72	.2519
−30	8.719	102.19	.2645	7.461	102.15	.2609	6.517	102.10	.2578	5.783	102.06	.2551
−20	8.927	103.54	.2676	7.640	103.51	.2640	6.673	103.47	.2610	5.923	103.43	.2582
−10	9.135	104.92	.2707	7.818	104.88	.2671	6.830	104.84	.2641	6.062	104.81	.2613
0	9.342	106.31	.2738	7.996	106.27	.2702	6.986	106.23	.2671	6.201	106.19	.2644
10	9.549	107.71	.2768	8.174	107.67	.2732	7.143	107.63	.2701	6.341	107.60	.2674
20	9.757	109.12	.2798	8.352	109.09	.2762	7.299	109.05	.2731	6.479	109.02	.2704
30	9.964	110.55	.2827	8.530	110.52	.2792	7.455	110.48	.2761	6.618	110.45	.2734
40	10.171	112.00	.2856	8.708	111.96	.2821	7.611	111.93	.2790	6.757	111.90	.2763
50	10.378	113.46	.2885	8.886	113.42	.2850	7.767	113.39	.2819	6.896	113.36	.2792
60	10.585	114.93	.2914	9.064	114.90	.2878	7.922	114.87	.2848	7.035	114.83	.2820
70	10.792	116.42	.2942	9.241	116.39	.2907	8.078	116.36	.2876	7.173	116.32	.2849
80	10.999	117.92	.2970	9.419	117.89	.2935	8.234	117.86	.2904	7.312	117.83	.2877
90	11.206	119.44	.2998	9.597	119.41	.2963	8.389	119.38	.2932	7.450	119.35	.2905
100	11.413	120.97	.3026	9.774	120.94	.2990	8.545	120.91	.2960	7.589	120.88	.2933
110	11.620	122.52	.3053	9.952	122.49	.3018	8.700	122.46	.2987	7.727	122.43	.2960
120	11.826	124.08	.3080	10.129	124.05	.3045	8.856	124.02	.3014	7.865	124.00	.2987
130	12.033	125.66	.3107	10.306	125.63	.3072	9.011	125.60	.3041	8.004	125.57	.3014
140	12.239	127.25	.3134	10.483	127.22	.3099	9.166	127.19	.3068	8.142	127.17	.3041
150	12.446	128.85	.3161	10.660	128.82	.3125	9.321	128.80	.3095	8.280	128.77	.3068
160	12.652	130.47	.3187	10.837	130.45	.3152	9.477	130.42	.3121	8.418	130.39	.3094
170	12.859	132.11	.3213	11.014	132.08	.3178	9.632	132.06	.3147	8.556	132.03	.3120
180	13.065	133.75	.3239	11.192	133.73	.3204	9.787	133.71	.3173	8.694	133.68	.3146
190	13.271	135.41	.3265	11.369	135.40	.3230	9.942	135.37	.3199	8.832	135.35	.3172
200	13.477	137.09	.3291	11.545	137.08	.3255	10.097	137.05	.3225	8.970	137.03	.3198
210	13.683	138.79	.3316	11.722	138.77	.3281	10.252	138.75	.3250	9.108	138.72	.3223
220	13.889	140.50	.3341	11.899	140.47	.3306	10.407	140.45	.3276	9.245	140.43	.3249
230	14.096	142.22	.3366	12.076	142.20	.3331	10.562	142.18	.3301	9.383	142.15	.3274
240				12.253	143.94	.3356	10.717	143.91	.3326	9.521	143.89	.3299
250										9.659	145.65	.3324

Temp F	Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp −55.8 F)			Pressure 11 psia Gage pressure 7.53 in. vac (Sat temp −52.4 F)			Pressure 12 psia Gage pressure 5.50 in. vac (Sat temp −49.2 F)			Pressure 14 psia Gage pressure 1.42 in. vac (Sat temp −43.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(4.870)*	*(98.68)*	*(0.2444)*	*(4.468)*	*(98.99)*	*(0.2432)*	*(4.112)*	*(99.37)*	*(0.2422)*	*(3.662)*	*(100.06)*	*(0.2404)*
−50	4.943	99.35	.2462	4.485	99.31	.2440						
−40	5.069	100.68	.2494	4.600	100.64	.2472	4.209	100.60	.2452	3.595	100.51	.2415
−30	5.195	102.02	.2526	4.715	101.98	.2504	4.315	101.94	.2484	3.685	101.86	.2447
−20	5.321	103.29	.2558	4.830	103.35	.2535	4.420	103.31	.2515	3.776	103.23	.2479
−10	5.447	104.77	.2589	4.944	104.73	.2567	4.525	104.69	.2546	3.867	104.62	.2510
0	5.573	106.16	.2619	5.059	106.12	.2597	4.630	106.08	.2577	3.957	106.01	.2541
10	5.699	107.56	.2650	5.173	107.53	.2628	4.735	107.49	.2607	4.048	107.42	.2571
20	5.824	108.98	.2680	5.288	108.95	.2658	4.840	108.91	.2637	4.138	108.84	.2601
30	5.950	110.40	.2709	5.402	110.38	.2687	4.945	110.35	.2667	4.228	110.28	.2631
40	6.075	111.86	.2739	5.516	111.83	.2716	5.050	111.79	.2696	4.318	111.73	.2660
50	6.200	113.32	.2767	5.630	113.99	.2745	5.155	113.26	.2725	4.408	113.19	.2689
60	6.325	114.80	.2796	5.743	114.77	.2774	5.260	114.74	.2754	4.499	114.67	.2718
70	6.450	116.29	.2824	5.857	116.26	.2802	5.364	116.23	.2782	4.588	116.17	.2746
80	6.575	117:80	.2853	5.971	117.77	.2831	5.468	117.74	.2810	4.678	117.68	.2775
90	6.700	119.32	.2881	6.085	119.29	.2859	5.573	119.26	.2838	4.768	119.20	.2803
100	6.834	120.85	.2908	6.199	120.82	.2886	5.677	120.80	.2866	4.857	120.74	.2830
110	6.949	122.40	.2936	6.312	122.37	.2914	5.781	122.35	.2894	4.947	122.29	.2858
120	7.073	123.97	.2963	6.425	123.94	.2941	5.885	123.91	.2921	5.036	123.86	.2885
130	7.198	125.55	.2990	6.539	125.59	.2968	5.989	125.49	.2948	5.126	125.44	.2912
140	7.322	127.14	.3017	6.652	127.11	.2995	6.093	127.08	.2975	5.215	127.03	.2939
150	7.447	128.74	.3043	6.766	128.72	.3021	6.197	128.69	.3001	5.304	128.64	.2966
160	7.571	130.37	.3070	6.879	130.34	.3048	6.301	130.32	.3028	5.394	130.27	.2992
170	7.696	132.01	.3096	6.992	131.98	.3074	6.405	131.96	.3054	5.483	131.91	.3018
180	7.820	133.66	.3122	7.105	133.63	.3100	6.509	133.61	.3080	5.573	133.56	.3044
190	7.945	135.32	.3148	7.218	135.30	.3126	6.613	135.27	.3106	5.661	135.23	.3070
200	8.069	137.00	.3173	7.331	136.98	.3152	6.717	136.96	.3132	5.750	136.91	.3096
210	8.193	138.70	.3199	7.444	138.68	.3177	6.820	138.65	.3157	5.840	138.61	.3121
220	8.317	140.41	.3224	7.557	140.39	.3202	6.924	140.36	.3182	5.929	140.32	.3147
230	8.441	142.13	.3250	7.670	142.11	.3228	7.028	142.09	.3208	6.018	142.05	.3172
240	7.565	143.87	.3275	7.783	143.85	.3253	7.131	143.83	.3233	6.107	143.79	.3197
250	8.689	145.63	.3300	7.896	145.60	.3278	7.234	145.58	.3258	6.196	145.54	.3222
260							7.338	147.35	.3282	6.285	147.31	.3247

Table 21. F-22 ($CHClF_2$) Monochlorodifluoromethane
Properties of Superheated Vapor (Continued)

Temp F	Pressure 16 psia Gage pressure 1.3 psig (Sat temp −38.3 F)			Pressure 18 psia Gage pressure 3.3 psig (Sat temp −33.7 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp −29.4 F)			Pressure 25 psia Gage pressure 10.3 psig (Sat temp −20.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*3.147*)	(*100.63*)	(*0.2390*)	(*2.820*)	(*101.19*)	(*0.2376*)	(*2.566*)	(*101.70*)	(*0.2365*)	(*2.074*)	(*102.79*)	(*0.2341*)
− 30	3.213	101.78	.2416	2.846	101.70	.2388						
− 20	3.293	103.16	.2447	2.917	103.08	.2420	2.616	103.00	.2394			
− 10	3.373	104.54	.2479	2.988	104.46	.2451	2.681	104.38	.2426	2.127	104.19	.2372
0	3.452	105.93	.2509	3.059	105.86	.2482	2.745	105.78	.2457	2.179	105.59	.2403
10	3.532	107.34	.2540	3.130	107.27	.2512	2.809	107.20	.2487	2.231	107.01	.2434
20	3.611	108.77	.2570	3.201	108.70	.2542	2.873	108.63	.2517	2.283	108.45	.2464
30	3.690	110.21	.2600	3.272	110.14	.2572	2.937	110.07	.2547	2.335	109.89	.2494
40	3.769	111.66	.2629	3.343	111.59	.2601	3.001	111.52	.2577	2.386	111.35	.2524
50	3.848	113.13	.2658	3.413	113.06	.2630	3.065	113.00	.2606	2.437	112.83	.2553
60	3.928	114.61	.2687	3.484	114.55	.2659	3.128	114.48	.2634	2.489	114.32	.2582
70	4.006	116.11	.2715	3.554	116.04	.2688	3.192	115.98	.2663	2.540	115.82	.2610
80	4.085	117.62	.2744	3.624	117.56	.2716	3.255	117.49	.2691	2.591	117.34	.2639
90	4.164	119.14	.2772	3.694	119.08	.2744	3.319	119.02	.2719	2.643	118.87	.2667
100	4.243	120.68	.2799	3.764	120.62	.2772	3.382	120.56	.2747	2.693	120.42	.2695
110	4.321	122.23	.2827	3.834	122.18	.2799	3.445	122.12	.2775	2.744	121.98	.2722
120	4.399	123.80	.2854	3.904	123.74	.2827	3.508	123.69	.2802	2.795	123.55	.2750
130	4.478	125.38	.2881	3.975	125.33	.2854	3.571	125.27	.2829	2.846	125.14	.2777
140	4.557	126.98	.2908	4.044	126.92	.2881	3.635	126.87	.2856	2.897	126.74	.2804
150	4.635	128.59	.2935	4.114	128.54	.2907	3.698	128.49	.2883	2.947	128.36	.2831
160	4.713	130.21	.2961	4.184	130.16	.2934	3.761	130.11	.2909	2.998	129.99	.2857
170	4.792	131.86	.2987	4.254	131.81	.2960	3.824	131.76	.2936	3.049	131.63	.2883
180	4.870	133.51	.3014	4.324	133.46	.2986	3.886	133.41	.2962	3.099	133.29	.2910
190	4.948	135.18	.3039	4.393	135.13	.3012	3.949	135.08	.2988	3.150	134.96	.2936
200	5.026	136.86	.3065	4.463	136.82	.3038	4.012	136.77	.3013	3.200	136.65	.2961
210	5.104	138.56	.3091	4.532	138.52	.3063	4.074	138.47	.3039	3.251	138.36	.2987
220	5.182	140.27	.3116	4.602	140.32	.3089	4.137	140.19	.3064	3.301	140.07	.3012
230	5.260	142.00	.3141	4.671	141.96	.3114	4.200	141.91	.3090	3.352	141.80	.3038
240	5.338	143.74	.3166	4.740	143.70	.3139	4.263	143.66	.3115	3.402	143.55	.3063
250	5.416	145.50	.3191	4.810	145.46	.3164	4.325	145.42	.3140	3.452	145.31	.3088
260	5.494	147.27	.3216	4.880	147.23	.3189	4.388	147.19	.3165	3.503	147.08	.3113
270	5.572	149.05	.3241	4.949	149.02	.3214	4.450	148.98	.3189	3.553	148.88	.3138
280							4.512	150.78	.3214	3.603	150.68	.3162
290										3.653	152.50	.3187

Temp F	Pressure 30 psia Gage pressure 15.3 psig (Sat temp −11.9 F)			Pressure 35 psia Gage pressure 20.3 psig (Sat temp −4.9 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 1.5 F)			Pressure 45 psia Gage pressure 30.3 psig (Sat temp 7.2 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*1.749*)	(*103.71*)	(*0.2322*)	(*1.613*)	(*104.49*)	(*0.2306*)	(*1.334*)	(*105.17*)	(*0.2293*)	(*1.193*)	(*105.81*)	(*0.2281*)
− 10	1.758	103.98	.2328									
0	1.802	105.39	.2359	1.532	105.19	.2321						
10	1.845	106.82	.2390	1.570	106.62	.2352	1.363	106.41	.2319	1.202	106.22	.2290
20	1.889	108.26	.2420	1.608	108.07	.2383	1.396	107.87	.2350	1.232	107.68	.2321
30	1.933	109.71	.2450	1.645	109.53	.2413	1.430	109.34	.2380	1.262	109.16	.2351
40	1.976	111.18	.2480	1.683	111.00	.2443	1.463	110.82	.2410	1.292	110.65	.2381
50	2.019	112.66	.2509	1.720	112.49	.2472	1.496	112.31	.2439	1.322	112.15	.2411
60	2.063	114.15	.2538	1.758	113.99	.2501	1.529	113.81	.2469	1.351	113.64	.2440
70	2.106	115.66	.2567	1.795	115.50	.2530	1.562	115.33	.2498	1.381	115.16	.2469
80	2.149	117.18	.2595	1.832	117.02	.2559	1.595	116.86	.2526	1.410	116.71	.2498
90	2.192	118.72	.2624	1.869	118.56	.2587	1.628	118.40	.2555	1.440	118.26	.2526
100	2.235	120.27	.2652	1.906	120.12	.2615	1.660	119.96	.2583	1.469	119.81	.2554
110	2.277	121.83	.2679	1.943	121.68	.2643	1.693	121.53	.2610	1.498	121.38	.2582
120	2.320	123.41	.2707	1.980	123.26	.2670	1.726	123.12	.2638	1.527	122.97	.2610
130	2.362	125.00	.2734	2.017	124.86	.2697	1.758	124.72	.2665	1.556	124.57	.2637
140	2.405	126.61	.2761	2.054	126.47	.2724	1.790	126.33	.2693	1.585	126.19	.2664
150	2.448	128.22	.2788	2.091	128.09	.2751	1.823	127.96	.2719	1.614	127.82	.2691
160	2.490	129.86	.2814	2.127	129.73	.2778	1.855	129.60	.2746	1.643	129.47	.2718
170	2.533	131.51	.2841	2.164	131.38	.2804	1.887	131.25	.2773	1.672	131.12	.2745
180	2.575	133.17	.2867	2.200	133.04	.2830	1.919	132.92	.2799	1.701	132.79	.2771
190	2.617	134.84	.2893	2.237	134.72	.2857	1.951	134.60	.2825	1.730	134.48	.2797
200	2.659	136.54	.2919	2.273	136.42	.2882	1.983	136.30	.2851	1.758	136.18	.2823
210	2.702	138.24	.2944	2.310	138.12	.2908	2.016	138.01	.2877	1.787	137.89	.2849
220	2.744	139.96	.2970	2.346	139.83	.2934	2.047	139.73	.2902	1.815	139.62	.2875
230	2.786	141.69	.2995	2.382	141.58	.2959	2.079	141.47	.2928	1.844	141.35	.2900
240	2.829	143.44	.3020	2.419	143.33	.2984	2.111	143.22	.2953	1.872	143.11	.2925
250	2.871	145.20	.3045	2.455	145.10	.3009	2.143	144.99	.2978	1.902	144.87	.2950
260	2.913	146.98	.3070	2.491	146.88	.3034	2.175	146.77	.3003	1.931	146.66	.2975
270	2.955	148.77	.3095	2.527	148.67	.3059	2.207	148.57	.3028	1.959	148.46	.3000
280	2.997	150.58	.3120	2.563	150.48	.3084	2.239	150.38	.3052	1.987	150.28	.3025
290	3.039	152.40	.3144	2.600	152.30	.3108	2.270	152.20	.3077	2.016	152.11	.3049
300				2.636	154.14	.3132	2.302	154.04	.3101	2.044	153.94	.3074
310							2.334	155.89	.3125	2.072	155.80	.3098

Table 21. F-22 (CHClF₂) Monochlorodifluoromethane—Properties of Superheated Vapor (Continued)

Temp F	Pressure 50 psia Gage pressure 35.3 psig (Sat temp 12.4 F)			Pressure 55 psia Gage pressure 40.3 psig (Sat temp 17.2 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 21.8 F)			Pressure 65 psia Gage pressure 50.3 psig (Sat temp 26.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	*(1.080)*	*(106.96)*	*(0.2271)*	*(.9851)*	*(106.85)*	*(0.2261)*	*(.9061)*	*(107.32)*	*(0.2253)*	*(.8398)*	*(107.73)*	*(0.2242)*
20	1.100	107.48	.2294	.9922	107.27	.2270						
30	1.128	108.96	.2325	1.0176	108.76	.2300	.9255	108.56	.2278	.8480	108.36	.2257
40	1.155	110.46	.2355	1.0425	110.27	.2331	.9491	110.07	.2308	.8700	109.88	.2288
50	1.182	111.96	.2384	1.0675	111.78	.2360	.9721	111.59	.2338	.8917	111.39	.2318
60	1.209	113.47	.2414	1.0921	113.28	.2390	.9950	113.10	.2368	.9130	112.91	.2347
70	1.236	114.99	.2443	1.1168	114.82	.2419	1.0180	114.64	.2397	.9343	114.46	.2377
80	1.262	116.54	.2472	1.1414	116.37	.2448	1.0406	116.20	.2426	.9557	116.02	.2406
90	1.289	118.10	.2500	1.1659	117.93	.2477	1.0634	117.77	.2455	.9763	117.60	.2435
100	1.316	119.66	.2528	1.1902	119.50	.2505	1.0858	119.35	.2483	.9973	119.18	.2463
110	1.342	121.23	.2556	1.2146	121.08	.2533	1.1082	120.92	.2511	1.0183	120.77	.2491
120	1.369	122.83	.2584	1.2386	122.68	.2561	1.1305	122.53	.2539	1.0390	122.38	.2519
130	1.395	124.43	.2612	1.2627	124.29	.2588	1.1529	124.14	.2567	1.0598	124.00	.2547
140	1.421	126.05	.2639	1.2870	125.91	.2616	1.1751	125.77	.2594	1.0804	125.63	.2574
150	1.448	127.68	.2666	1.3110	127.54	.2643	1.1974	127.41	.2621	1.1010	127.27	.2602
160	1.474	129.33	.2693	1.3349	129.20	.2670	1.2193	129.06	.2648	1.1217	128.93	.2629
170	1.500	130.99	.2719	1.3590	130.86	.2696	1.2415	130.73	.2675	1.1421	130.60	.2655
180	1.526	132.67	.2764	1.3827	132.54	.2723	1.2634	132.41	.2702	1.1627	132.28	.2682
190	1.552	134.35	.2772	1.4065	134.23	.2749	1.2853	134.10	.2728	1.1829	133.98	.2708
200	1.578	136.05	.2798	1.4301	135.93	.2775	1.3071	135.81	.2754	1.2031	135.68	.2734
210	1.604	137.77	.2824	1.4539	137.65	.2801	1.3290	137.53	.2780	1.2234	137.41	.2760
220	1.630	139.50	.2849	1.4774	139.38	.2827	1.3509	139.26	.2805	1.2437	139.15	.2786
230	1.655	141.24	.2875	1.5011	141.12	.2852	1.3726	141.01	.2831	1.2639	140.89	.2812
240	1.681	143.00	.2900	1.5246	142.88	.2877	1.3943	142.77	.2856	1.2841	142.66	.2837
250	1.707	144.76	.2925	1.5481	144.66	.2902	1.4159	144.55	.2882	1.3041	144.43	.2862
260	1.732	146.55	.2950	1.5716	146.44	.2927	1.4375	146.33	.2907	1.3240	146.23	.2887
270	1.758	148.35	.2975	1.5950	148.25	.2952	1.4591	148.14	.2932	1.3441	148.04	.2912
280	1.784	150.18	.3000	1.6184	150.07	.2977	1.4806	149.97	.2956	1.3641	149.87	.2937
290	1.810	152.01	.3024	1.6417	151.91	.3002	1.5021	151.81	.2981	1.3840	151.71	.2962
300	1.835	153.85	.3049	1.6651	153.75	.3026	1.5235	153.65	.3005	1.4038	153.55	.2986
310	1.861	155.70	.3073	1.6883	155.61	.3051	1.5451	155.51	.3030	1.4237	155.41	.3011
320	1.886	157.58	.3097	1.7116	157.48	.3075	1.5665	157.39	.3054	1.4436	157.29	.3035
330							1.5879	159.27	.3078	1.4634	159.18	.3059

Temp F	Pressure 70 psia Gage pressure 55.3 psig (Sat temp 30.0 F)			Pressure 75 psia Gage pressure 60.3 psig (Sat temp 33.8 F)			Pressure 80 psia Gage pressure 65.3 psig (Sat temp 37.4 F)			Pressure 90 psia Gage pressure 75.3 psig (Sat temp 44.2 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	*(.7816)*	*(108.13)*	*(0.2235)*	*(.7301)*	*(108.31)*	*(0.2231)*	*(.6856)*	*(108.86)*	*(0.2224)*	*(.6109)*	*(109.48)*	*(0.2212)*
40	.8023	109.67	.2268	.7421	109.47	.2249	.6904	109.26	.2232			
50	.8224	111.19	.2298	.7615	111.00	.2280	.7087	110.80	.2263	.6207	110.39	.2230
60	.8424	112.71	.2328	.7807	112.52	.2310	.7269	112.32	.2293	.6372	111.92	.2261
70	.8625	114.27	.2358	.7995	114.09	.2340	.7446	113.90	.2323	.6535	113.51	.2291
80	.8823	115.84	.2387	.8182	115.67	.2369	.7625	115.48	.2352	.6697	115.12	.2321
90	.9019	117.43	.2416	.8368	117.26	.2398	.7801	117.09	.2381	.6853	116.73	.2350
100	.9216	119.02	.2444	.8553	118.86	.2427	.7978	118.69	.2410	.7018	118.35	.2379
110	.9411	120.61	.2473	.8739	120.45	.2455	.8151	120.29	.2439	.7175	119.96	.2408
120	.9605	121.23	.2501	.8923	122.07	.2483	.8327	121.91	.2467	.7334	121.59	.2436
130	.9800	123.85	.2529	.9107	123.70	.2511	.8500	123.55	.2495	.7489	123.23	.2464
140	.9991	125.48	.2556	.9289	125.34	.2539	.8673	125.19	.2522	.7645	124.89	.2492
150	1.0187	127.12	.2583	.9470	126.99	.2566	.8844	126.84	.2550	.7800	126.55	.2520
160	1.0379	128.79	.2610	.9650	128.65	.2593	.9014	128.51	.2577	.7954	128.23	.2547
170	1.0569	130.47	.2637	.9831	130.33	.2620	.9184	130.19	.2604	.8108	129.92	.2574
180	1.0762	132.15	.2664	1.0012	132.02	.2647	.9356	131.89	.2630	.8262	131.62	.2601
190	1.0951	133.85	.2690	1.0188	133.72	.2673	.9524	133.59	.2657	.8411	133.34	.2627
200	1.1139	135.56	.2716	1.0366	135.44	.2699	.9689	135.31	.2683	.8561	135.06	.2654
210	1.1330	137.29	.2742	1.0544	137.17	.2725	.9857	137.04	.2709	.8711	136.80	.2680
220	1.1519	139.03	.2768	1.0722	138.91	.2751	1.0025	138.79	.2735	.8863	138.55	.2706
230	1.1707	140.78	.2794	1.0898	140.66	.2777	1.0192	140.54	.2761	.9013	140.31	.2732
240	1.1894	142.54	.2819	1.1075	142.43	.2802	1.0358	142.32	.2787	.9163	142.08	.2757
250	1.2082	144.32	.2844	1.1250	144.21	.2828	1.0524	144.10	.2812	.9312	143.87	.2783
260	1.2268	146.12	.2870	1.1426	146.01	.2853	1.0690	145.90	.2837	.9460	145.67	.2808
270	1.2456	147.93	.2895	1.1601	147.82	.2878	1.0854	147.72	.2862	.9608	147.50	.2833
280	1.2642	149.76	.2919	1.1776	149.66	.2903	1.1019	149.55	.2887	.9756	149.34	.2858
290	1.2829	151.60	.2944	1.1949	151.50	.2927	1.1183	151.40	.2912	.9902	151.19	.2883
300	1.3013	153.45	.2969	1.2122	153.35	.2952	1.1343	153.25	.2936	1.0048	153.05	.2908
310	1.3198	155.31	.2993	1.2296	155.22	.2976	1.1507	155.12	.2961	1.0194	154.92	.2932
320	1.3384	157.20	.3017	1.2481	157.10	.3001	1.1671	157.00	.2985	1.0341	156.81	.2957
330	1.3569	159.09	.3041	1.2655	158.99	.3025	1.1834	158.90	.3009	1.0487	158.71	.2981
340	1.3753	160.99	.3066	1.2829	160.90	.3049	1.1996	160.80	.3033	1.0633	160.62	.3005
350										1.0779	162.54	.3029

Table 21. F-22 (CHClF$_2$) Monochlorodifluoromethane—Properties of Superheated Vapor (Continued)

Temp F	Pressure 100 psia Gage pressure 85.3 psig (Sat temp 50.4 F)			Pressure 110 psia Gage pressure 95.3 psig (Sat temp 56.1 F)			Pressure 120 psia Gage pressure 105.3 psig (Sat temp 61.5 F)			Pressure 130 psia Gage pressure 115.3 psig (Sat temp 66.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(.5505)*	*(110.03)*	*(0.2202)*	*(.5005)*	*(110.50)*	*(0.2192)*	*(.4586)*	*(110.91)*	*(0.2183)*	*(.4228)*	*(111.27)*	*(0.2178)*
60	.5651	111.51	.2232	.5061	111.10	.2204						
70	.5802	113.11	.2262	.5202	112.70	.2235	.4698	112.28	.2210	.4272	111.84	.2187
80	.5953	114.74	.2292	.5343	114.35	.2266	.4833	113.95	.2241	.4398	113.53	.2217
90	.6101	116.37	.2322	.5480	115.99	.2296	.4961	115.60	.2271	.4520	115.20	.2248
100	.6249	118.00	.2351	.5617	117.64	.2325	.5089	117.27	.2301	.4641	116.88	.2278
110	.6394	119.62	.2380	.5753	119.27	.2354	.5215	118.91	.2330	.4761	118.54	.2307
120	.6539	121.26	.2408	.5886	120.93	.2383	.5343	120.58	.2359	.4879	120.23	.2337
130	.6682	122.92	.2437	.6020	122.59	.2411	.5466	122.26	.2388	.4996	121.92	.2365
140	.6824	124.58	.2465	.6151	124.27	.2440	.5589	123.94	.2416	.5112	123.61	.2394
150	.6965	126.25	.2492	.6281	125.94	.2467	.5711	125.63	.2444	.5227	125.31	.2422
160	.7107	127.94	.2520	.6412	127.64	.2495	.5832	127.34	.2472	.5341	127.03	.2450
170	.7246	129.64	.2547	.6541	129.35	.2522	.5952	129.06	.2499	.5454	128.76	.2478
180	.7386	131.35	.2574	.6669	131.07	.2549	.6072	130.79	.2526	.5566	130.50	.2505
190	.7523	133.07	.2601	.6796	132.80	.2576	.6190	132.53	.2553	.5677	132.25	.2532
200	.7660	134.80	.2627	.6922	134.54	.2603	.6307	134.27	.2580	.5786	134.01	.2559
210	.7796	136.55	.2654	.7048	136.29	.2629	.6424	136.03	.2607	.5895	135.77	.2586
220	.7933	138.30	.2680	.7174	138.05	.2655	.6541	137.80	.2633	.6005	137.55	.2612
230	.8071	140.07	.2706	.7299	139.82	.2681	.6657	139.58	.2659	.6114	139.33	.2638
240	.8206	141.85	.2731	.7424	141.61	.2707	.6772	141.37	.2685	.6221	141.13	.2664
250	.8343	143.64	.2757	.7550	143.40	.2733	.6888	143.17	.2711	.6330	142.94	.2690
260	.8477	145.45	.2782	.7673	145.22	.2758	.7003	144.99	.2736	.6437	144.76	.2716
270	.8612	147.28	.2807	.7796	147.06	.2784	.7118	146.84	.2762	.5543	146.61	.2741
280	.8746	149.13	.2832	.7919	148.91	.2809	.7231	148.69	.2787	.6649	148.84	.2766
290	.8879	150.98	.2857	.8041	150.77	.2834	.7344	150.59	.2812	.6754	150.34	.2791
300	.9011	152.84	.2882	.8162	152.64	.2858	.7456	152.43	.2837	.6859	152.22	.2816
310	.9144	154.72	.2906	.8284	154.52	.2883	.7569	154.32	.2861	.6965	154.12	.2841
320	.9277	156.62	.2931	.8406	156.42	.2907	.7681	156.22	.2886	.7069	156.03	.2866
330	.9410	158.52	.2955	.8529	158.33	.2932	.7793	158.14	.2910	.7174	157.94	.2890
340	.9541	160.43	.2979	.8650	160.25	.2956	.7906	160.06	.2934	.7277	159.87	.2914
350	.9674	162.36	.3003	.8771	162.18	.2980	.8018	162.00	.2959	.7381	161.81	.2939
360	.9806	164.31	.3027	.8891	164.13	.3004	.8129	163.95	.2983	.7485	163.77	.2963
370							.8240	165.92	.3007	.7588	165.74	.2987

Temp F	Pressure 140 psia Gage pressure 125.3 psig (Sat temp 71.3 F)			Pressure 150 psia Gage pressure 135.3 psig (Sat temp 75.8 F)			Pressure 160 psia Gage pressure 145.3 psig (Sat temp 80.1 F)			Pressure 180 psia Gage pressure 165.3 psig (Sat temp 88.2 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(.3919)*	*(111.60)*	*(0.2167)*	*(.3648)*	*(111.89)*	*(0.2160)*	*(.3411)*	*(112.06)*	*(0.2152)*	*(.3012)*	*(112.60)*	*(0.2138)*
80	.4022	113.08	.2195	.3695	112.61	.2173						
90	.4141	114.79	.2226	.3810	114.35	.2205	.3518	113.90	.2184	.3031	112.93	.2145
100	.4257	116.48	.2256	.3922	116.07	.2235	.3627	115.64	.2215	.3135	114.73	.2177
110	.4369	118.16	.2286	.4030	117.77	.2265	.3733	117.36	.2246	.3236	116.51	.2208
120	.4483	119.86	.2315	.4139	119.49	.2296	.3838	119.10	.2276	.3331	118.29	.2239
130	.4593	121.57	.2344	.4245	121.21	.2325	.3939	120.84	.2306	.3427	120.07	.2269
140	.4704	123.27	.2373	.4349	122.93	.2354	.4039	122.58	.2335	.3520	121.85	.2299
150	.4812	124.99	.2402	.4453	124.66	.2382	.4138	124.33	.2364	.3611	123.63	.2329
160	.4921	126.72	.2430	.4556	126.40	.2411	.4236	126.08	.2392	.3701	125.41	.2358
170	.5027	128.46	.2458	.4657	128.15	.2439	.4332	127.84	.2420	.3791	127.19	.2386
180	.5132	130.20	.2485	.4757	129.91	.2466	.4429	129.60	.2448	.3879	128.98	.2415
190	.5236	131.96	.2512	.4856	131.67	.2494	.4522	131.38	.2476	.3966	130.78	.2443
200	.5340	133.73	.2539	.4953	133.45	.2521	.4615	133.17	.2503	.4051	132.59	.2470
210	.5444	135.50	.2566	.5051	135.24	.2548	.4709	134.96	.2530	.4137	134.41	.2498
220	.5546	137.29	.2593	.5149	137.03	.2574	.4801	136.76	.2557	.4221	136.23	.2525
230	.5649	139.08	.2619	.5245	138.83	.2601	.4892	138.57	.2583	.4303	138.06	.2551
240	.5749	140.89	.2645	.5341	140.64	.2624	.4983	140.40	.2610	.4386	139.90	.2578
250	.5850	142.70	.2671	.5436	142.47	.2653	.5073	142.33	.2636	.4468	141.75	.2604
260	.5951	144.54	.2697	.5530	144.31	.2679	.5162	144.08	.2662	.4550	143.61	.2630
270	.6051	146.39	.2722	.5625	146.17	.2704	.5251	145.94	.2687	.4630	145.49	.2656
280	.6150	148.26	.2747	.5718	148.04	.2730	.5340	147.82	.2713	.4711	147.38	.2682
290	.6248	150.13	.2773	.5811	149.92	.2755	.5428	149.70	.2738	.4791	149.27	.2707
300	.6347	152.02	.2798	.5903	151.81	.2780	.5516	151.60	.2763	.4870	151.17	.2732
310	.6446	153.91	.2822	.5997	153.71	.2805	.5604	153.51	.2788	.4951	153.09	.2757
320	.6544	155.83	.2847	.6090	155.63	.2829	.5692	155.43	.2813	.5029	155.03	.2782
330	.6641	157.75	.2871	.6181	157.56	.2854	.5777	157.36	.2838	.5106	156.97	.2807
340	.6739	159.68	.2896	.6278	159.59	.2878	.5864	159.31	.2862	.5184	158.92	.2832
350	.6835	161.63	.2920	.6364	161.45	.2903	.5950	161.26	.2886	.5263	160.89	.2856
360	.6933	163.59	.2944	.6454	163.41	.2927	.6036	163.23	.2911	.5340	163.87	.2881
370	.7029	165.57	.2968	.6545	165.39	.2951	.6121	165.21	.2935	.5417	164.87	.2905
380	.7125	167.56	.2992	.6635	167.39	.2975	.6207	167.22	.2959	.5493	166.87	.2929
390							.6292	169.22	.2982	.5570	168.87	.2953

Table 21. F-22 (CHClF$_2$) Monochlorodifluoromethane—Properties of Superheated Vapor (Concluded)

Temp F	Pressure 200 psia Gage pressure 185.3 psig (Sat temp 95.6 F)			Pressure 220 psia Gage pressure 205.3 psig (Sat temp 102.5 F)			Pressure 240 psia Gage pressure 225.3 psig (Sat temp 108.9 F)			Pressure 260 psia Gage pressure 245.3 psig (Sat temp 115.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(.2690)*	*(112.03)*	*(0.2124)*	*(.2425)*	*(113.17)*	*(0.2112)*	*(.2202)*	*(113.28)*	*(0.2099)*	*(.2013)*	*(113.42)*	*(0.2084)*
100	.2734	113.75	.2139									
110	.2829	115.61	.2172	.2498	114.62	.2138	.2207	113.51	.2103			
120	.2922	117.42	.2204	.2586	116.49	.2171	.2305	115.48	.2137	.2066	114.45	.2106
130	.3013	119.25	.2236	.2674	118.37	.2203	.2388	117.43	.2171	.2147	116.47	.2141
140	.3102	121.06	.2266	.2759	120.23	.2235	.2471	119.36	.2204	.2227	118.45	.2174
150	.3189	122.89	.2296	.2843	122.11	.2265	.2551	121.29	.2236	.2303	120.44	.2207
160	.3274	124.71	.2326	.2923	123.96	.2295	.2629	123.18	.2266	.2379	122.36	.2238
170	.3358	126.52	.2355	.3004	125.81	.2325	.2705	125.06	.2297	.2454	124.29	.2270
180	.3439	128.34	.2384	.3079	127.66	.2354	.2779	126.95	.2327	.2526	126.22	.2300
190	.3521	130.17	.2412	.3155	129.52	.2383	.2851	128.85	.2356	.2596	128.16	.2330
200	.3599	132.00	.2440	.3229	131.39	.2411	.2922	130.76	.2385	.2663	130.10	.2359
210	.3679	133.84	.2467	.3304	133.25	.2439	.2992	132.64	.2413	.2728	132.01	.2388
220	.3756	135.68	.2495	.3377	135.12	.2467	.3061	134.53	.2441	.2793	133.92	.2416
230	.3834	137.53	.2522	.3450	136.99	.2494	.3128	136.42	.2469	.2857	135.84	.2444
240	.3909	139.39	.2548	.3519	138.87	.2521	.3196	138.22	.2496	.2921	137.76	.2472
250	.3985	141.25	.2575	.3590	140.76	.2548	.3262	140.24	.2523	.2984	139.70	.2499
260	.4060	143.13	.2601	.3660	142.64	.2575	.3326	142.14	.2550	.3045	141.62	.2526
270	.4134	145.02	.2627	.3729	144.55	.2601	.3392	144.06	.2576	.3107	143.56	.2553
280	.4209	146.92	.2653	.3798	146.46	.2627	.3456	145.99	.2603	.3168	145.51	.2580
290	.4281	148.83	.2679	.3866	148.38	.2653	.3520	147.93	.2629	.3229	147.46	.2606
300	.4351	150.75	.2704	.3933	150.31	.2678	.3582	149.87	.2654	.3287	149.43	.2632
310	.4428	152.68	.2729	.4001	152.25	.2704	.3645	151.82	.2680	.3346	151.39	.2658
320	.4499	154.62	.2754	.4067	154.21	.2729	.3707	153.79	.2705	.3404	153.37	.2683
330	.4572	156.57	.2779	.4133	156.17	.2754	.3769	155.77	.2731	.3462	155.36	.2709
340	.4642	158.54	.2804	.4199	158.15	.2779	.3830	157.75	.2755	.3520	157.36	.2734
350	.4713	160.51	.2829	.4264	160.13	.2804	.3892	159.75	.2780	.3577	159.36	.2759
360	.4784	162.50	.2853	.4330	162.13	.2828	.3952	161.75	.2805	.3633	161.38	.2783
370	.4855	164.50	.2877	.4394	164.14	.2852	.4012	163.77	.2829	.3690	163.41	.2808
380	.4924	166.51	.2902	.4459	166.16	.2877	.4072	165.81	.2854	.3746	165.46	.2832
390	.4994	168.53	.2926	.4524	168.19	.2901	.4132	167.85	.2878	.3802	167.51	.2857
400	.5064	170.57	.2949	.4588	170.24	.2925	.4192	169.90	.2902	.3858	169.57	.2881
410				.4652	172.31	.2949	.4251	171.98	.2926	.3913	171.66	.2905
420										.3968	173.77	.2929

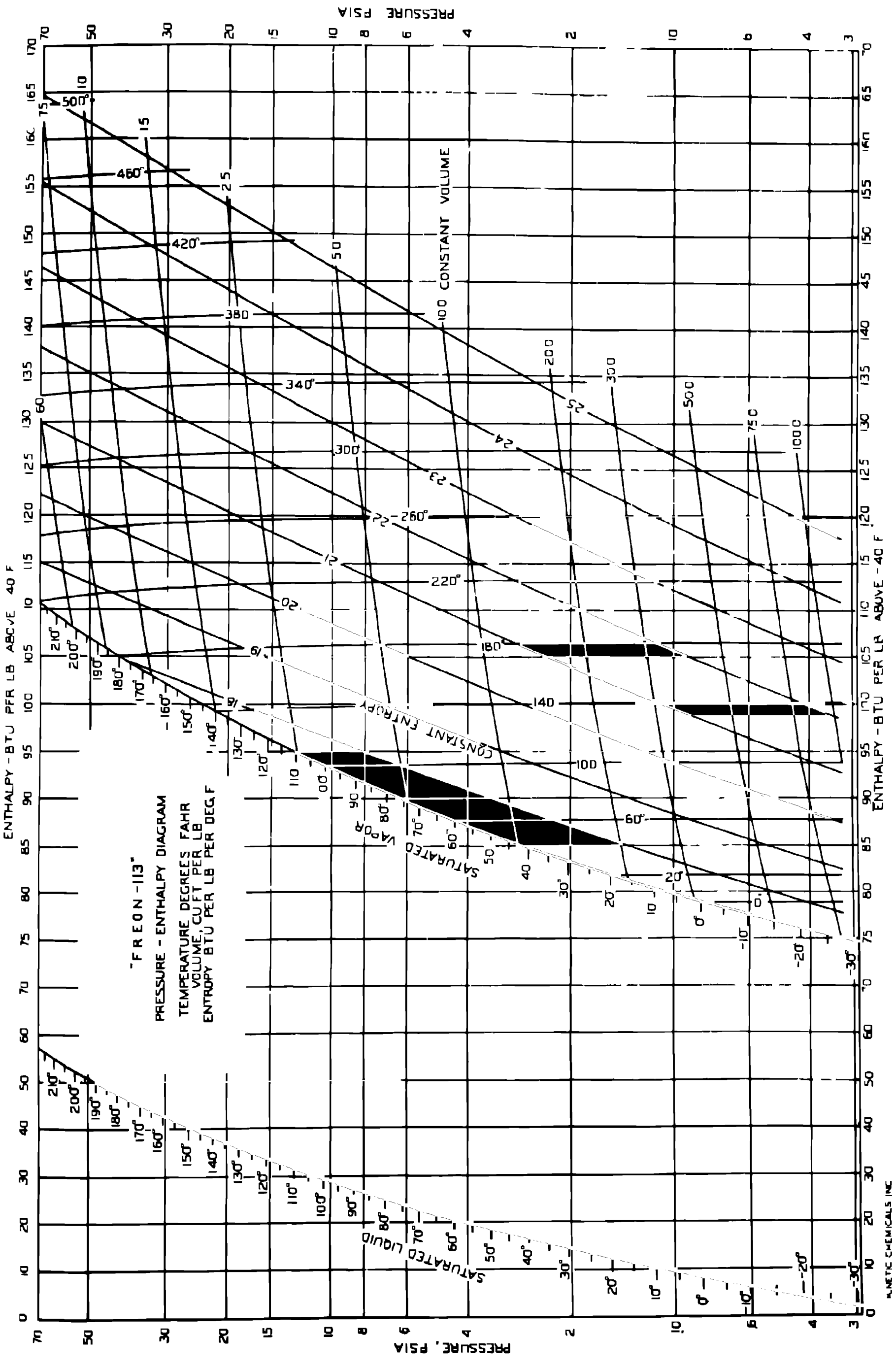

"FREON-113"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE DEGREES FAHR
VOLUME, CU FT PER LB
ENTROPY BTU PER LB PER DEG F
ENTHALPY - BTU PER LB ABOVE -40 F
PRESSURE, PSIA
CONSTANT VOLUME
CONSTANT ENTROPY
SATURATED VAPOR
SATURATED LIQUID
0 10 20 30 40 50 60 70 75 80 85 90 95 100 105 110 115 120 125 130 135 140 145 150 155 160 165 170
3 4 6 8 10 15 20 30 50 70
500° 460° 420° 380 340° 300° 260° 220° 180° 140 100 60° 20° 0° -20° -30°
10 15 25 50 100 200 300 500 750 1000
.16 .17 .18 .19 .20 .21 .22 .23 .24 .25
KINETIC CHEMICALS INC

Chart 11

Table 22. F-113 (CCl_2F-CCl_2F_2) Trichlorotrifluoroethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−30	0.2987	29.31*	0.00947	82.26	1.97	74.65	0.0047	0.1738
−20	.4288	29.05*	.00953	58.61	3.96	76.05	.0092	.1732
−10	.6046	28.69*	.00959	42.48	5.96	77.47	.0137	.1728
0	0.8377	28.21*	0.00966	31.31	7.98	78.89	0.0182	0.1725
2	.8924	28.10*	.00967	29.52	8.38	79.18	.0190	.1724
4	.9503	27.99*	.00968	27.84	8.78	79.46	.0199	.1724
5†	.9802	27.92*	.00969	27.04	8.98	79.60	.0203	.1723
6	1.011	27.86*	.00970	26.27	9.19	79.75	.0208	.1723
8	1.075	27.73*	.00971	24.81	9.59	80.03	.0216	.1723
10	1.142	27.60*	0.00972	23.45	10.00	80.32	0.0225	0.1723
12	1.213	27.45*	.00974	22.17	10.41	80.61	.0234	.1722
14	1.288	27.30*	.00975	20.97	10.81	80.89	.0242	.1722
16	1.366	27.14*	.00977	19.84	11.22	81.18	.0251	.1722
18	1.448	26.97*	.00978	18.79	11.62	81.46	.0259	.1722
20	1.534	26.80*	0.00979	17.81	12.03	81.75	0.0268	0.1722
22	1.624	26.61*	.00981	16.89	12.44	82.04	.0276	.1721
24	1.719	26.42*	.00982	16.02	12.85	82.33	.0285	.1721
26	1.818	26.22*	.00984	15.20	13.26	82.62	.0293	.1722
28	1.922	26.01*	.00985	14.43	13.67	82.91	.0302	.1722
30	2.031	25.79*	0.00987	13.71	14.08	83.20	0.0310	0.1722
32	2.145	25.55*	.00988	13.03	14.49	83.49	.0318	.1722
34	2.264	25.31*	.00990	12.39	14.91	83.78	.0327	.1722
36	2.388	25.06*	.00991	11.79	15.32	84.07	.0335	.1722
38	2.519	24.79*	.00993	11.22	15.74	84.36	.0343	.1722
40	2.655	24.52*	0.00994	10.68	16.16	84.65	0.0352	0.1723
42	2.797	24.23*	.00996	10.18	16.57	84.94	.0360	.1723
44	2.944	23.93*	.00997	9.703	16.99	85.24	.0368	.1723
46	3.098	23.61*	.00999	9.253	17.41	85.53	.0377	.1724
48	3.258	23.29*	.01000	8.830	17.82	85.82	.0385	.1724
50	3.427	22.94*	0.01002	8.426	18.24	86.11	0.0393	0.1725
52	3.602	22.59*	.01003	8.044	18.66	86.40	.0401	.1726
54	3.784	22.22*	.01005	7.682	19.08	86.69	.0410	.1726
56	3.973	21.83*	.01006	7.342	19.50	86.98	.0418	.1727
58	4.170	21.43*	.01008	7.018	19.93	87.28	.0426	.1727
60	4.374	21.02*	0.01010	6.713	20.35	87.57	0.0434	0.1728
62	4.586	20.59*	.01011	6.424	20.77	87.86	.0442	.1729
64	4.807	20.14*	.01013	6.149	21.19	88.15	.0450	.1729
66	5.036	19.67*	.01015	5.889	21.62	88.45	.0459	.1730
68	5.275	19.18*	.01016	5.640	22.05	88.74	.0467	.1731
70	5.523	18.68*	0.01018	5.404	22.48	89.04	0.0475	0.1731
72	5.780	18.16*	.01019	5.180	22.90	89.33	.0483	.1732
74	6.042	17.62*	.01021	4.971	23.33	89.62	.0491	.1733
76	6.320	17.06*	.01023	4.769	23.76	89.92	.0499	.1734
78	6.607	16.47*	.01025	4.574	24.19	90.21	.0507	.1735
80	6.902	15.87*	0.01026	4.392	24.63	90.51	0.0515	0.1736
82	7.208	15.25*	.01028	4.218	25.06	90.80	.0523	.1737
84	7.527	14.60*	.01030	4.051	25.49	91.09	.0531	.1738
86†	7.856	13.93*	.01031	3.893	25.93	91.39	.0539	.1739
88	8.194	13.24*	.01033	3.742	26.36	91.68	.0547	.1740
90	8.545	12.53*	0.01035	3.600	26.80	91.98	0.0555	0.1741
92	8.908	11.79*	.01037	3.463	27.24	92.28	.0563	.1742
94	9.281	11.03*	.01039	3.333	27.67	92.57	.0571	.1743
96	9.668	10.24*	.01040	3.208	28.11	92.86	.0578	.1744
98	10.07	9.42*	.01042	3.089	28.55	93.15	.0586	.1745
100	10.48	8.59*	0.01044	2.976	28.99	93.45	0.0594	0.1746
120	15.40	0.70	.01063	2.078	33.48	96.41	.0673	.1758
140	21.93	7.23	.01083	1.491	38.05	99.36	.0750	.1773
160	30.44	15.74	.01105	1.094	42.74	102.29	.0827	.1788
180	41.22	26.52	.01128	0.8193	47.53	105.19	.0903	.1804
200	54.66	39.96	0.01153	0.6241	52.45	108.07	0.0978	0.1821

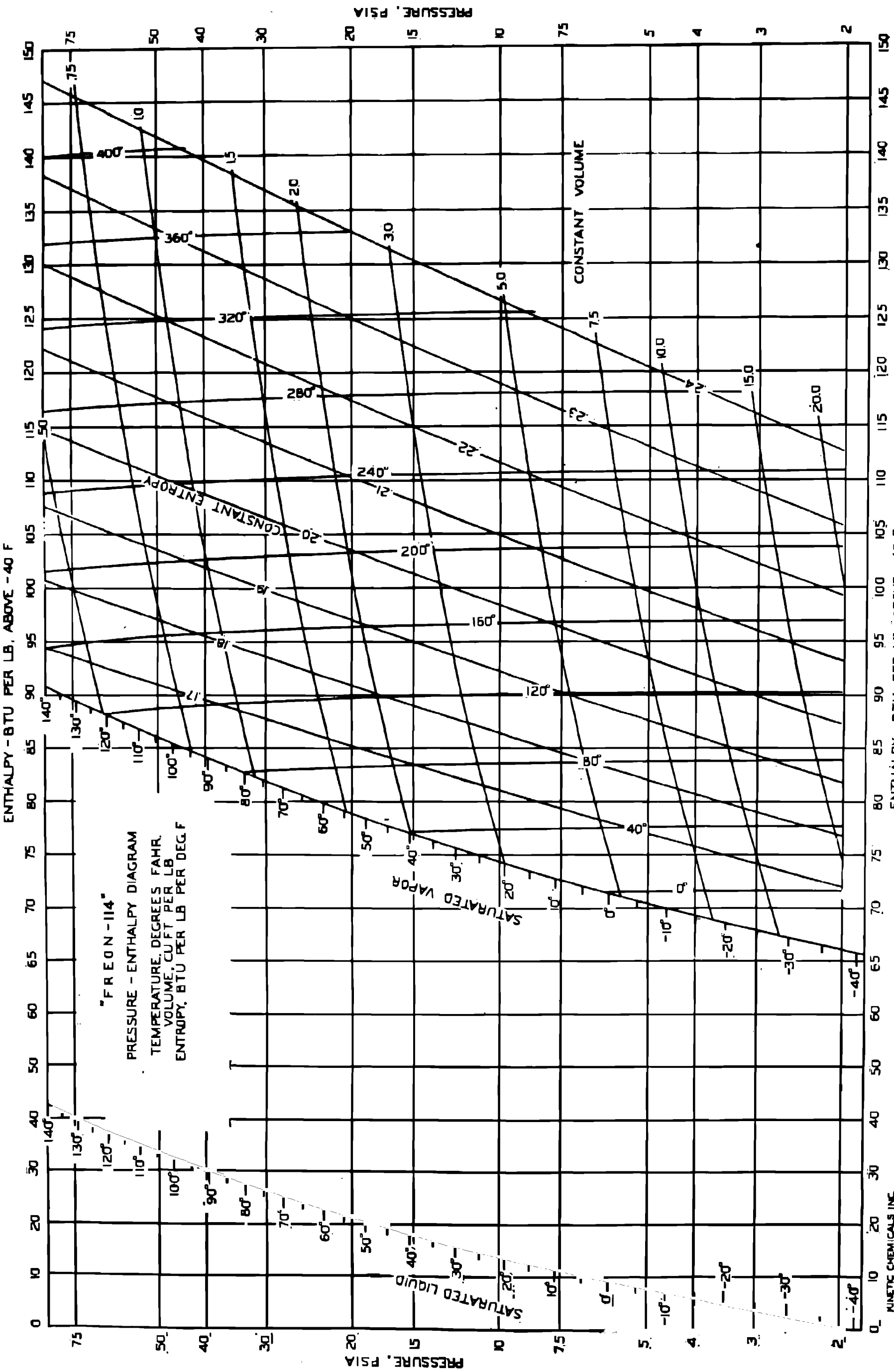
* Inches of mercury below one atmosphere.
† Standard cycle temperatures.

Table 23. F-113 (CCl_2F-$CClF_2$) Trichlorotrifluoroethane—Properties of Superheated Vapor

Temp F	Pressure 1.0 psia Gage pressure 27.88 in. vac (Sat temp 5.6 F)			Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp 29.4 F)			Pressure 3.0 psia Gage pressure 23.81 in. vac (Sat temp 44.7 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp 56.3 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(26.64)*	*(79.69)*	*(0.1723)*	*(13.91)*	*(83.12)*	*(0.1722)*	*(9.537)*	*(85.34)*	*(0.1723)*	*(7.297)*	*(87.03)*	*(0.1727)*
50	29.10	86.20	0.1857	14.50	86.16	0.1783	9.637	86.12	0.1739			
60	29.67	87.70	.1886	14.79	87.66	.1812	9.831	87.62	.1769	7.351	87.58	0.1738
70	30.25	89.31	.1915	15.08	89.17	.1841	10.02	89.12	.1797	7.497	89.10	.1767
80	30.82	90.73	.1943	15.37	90.69	.1869	10.22	90.66	.1826	7.642	90.62	.1795
90	31.39	92.27	.1972	15.66	92.23	.1898	10.41	92.19	.1854	7.787	92.15	.1823
100	31.97	93.81	0.2000	15.95	93.77	0.1926	10.60	93.74	0.1882	7.932	93.70	0.1851
110	32.56	95.37	.2027	16.24	95.33	.1953	10.80	95.30	.1910	8.077	95.26	.1879
120	33.14	96.94	.2054	16.53	96.90	.1980	10.99	96.87	.1937	8.222	96.83	.1906
130	33.71	98.52	.2081	16.82	98.49	.2007	11.18	98.46	.1964	8.367	98.42	.1933
140	34.28	100.11	.2108	17.11	100.09	.2034	11.38	100.05	.1991	8.522	100.01	.1960
150	34.85	101.72	0.2135	17.39	101.69	0.2061	11.57	101.65	0.2017	8.667	101.62	0.1986
160	35.42	103.33	.2161	17.68	103.31	.2087	11.76	103.27	.2044	8.812	103.23	.2003
170	35.99	104.96	.2187	17.96	104.93	.2113	11.95	104.90	.2070	8.957	104.86	.2039
180	36.57	106.60	.2213	17.25	106.57	.2139	12.15	106.54	.2095	9.102	106.51	.2065
190	37.15	108.25	.2238	18.54	108.22	.2164	12.34	108.19	.2121	9.247	108.16	.2091
200	37.73	109.91	0.2264	18.83	109.88	0.2190	12.53	109.85	0.2147	9.387	109.82	0.2116
210	38.30	111.59	.2289	19.12	111.56	.2215	12.72	111.53	.2172	9.532	111.50	.2141
220	38.88	113.28	.2314	19.40	113.24	.2240	12.92	113.22	.2197	9.677	113.19	.2166
230	39.46	114.97	.2339	19.69	114.94	.2265	13.11	114.92	.2222	9.822	114.88	.2191
240	40.03	116.68	.2363	19.97	116.65	.2290	13.30	116.63	.2246	9.967	116.59	.2216

Temp F	Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp 73.7 F)			Pressure 8.0 psia Gage pressure 13.64 in. vac (Sat temp 86.9 F)			Pressure 10.0 psia Gage pressure 9.57 in. vac (Sat temp 97.6 F)			Pressure 12 psia Gage pressure 5.50 in. vac (Sat temp 106.8 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(5.004)*	*(89.58)*	*(0.1733)*	*(3.828)*	*(91.52)*	*(0.1739)*	*(3.110)*	*(93.11)*	*(0.1746)*	*(2.624)*	*(94.46)*	*(0.1750)*
100	5.260	93.62	0.1807	3.924	93.56	0.1776	3.124	93.48	0.1751			
110	5.357	95.18	.1835	3.999	95.12	.1804	3.183	95.04	.1779	2.640	94.97	0.1759
120	5.454	96.76	.1862	4.073	96.69	.1831	3.242	96.62	.1806	2.689	96.55	.1786
130	5.551	98.35	.1889	4.147	98.28	.1858	3.301	98.21	.1834	2.738	98.14	.1813
140	5.649	99.95	.1916	4.221	99.88	.1885	3.361	99.81	.1860	2.788	99.73	.1840
150	5.746	101.56	0.1943	4.295	101.48	0.1912	3.420	101.41	0.1887	2.838	101.34	0.1867
160	5.843	103.17	.1969	4.369	103.10	.1938	3.480	103.03	.1914	2.888	102.97	.1894
170	5.940	104.80	.1995	4.442	104.74	.1964	3.539	104.66	.1940	2.938	104.60	.1920
180	6.037	106.45	.2021	4.515	106.38	.1990	3.599	106.31	.1966	2.987	106.25	.1964
190	6.134	108.10	.2047	4.588	108.04	.2016	3.658	107.96	.1991	3.037	107.90	.1972
200	6.233	109.77	0.2072	4.661	109.71	0.2041	3.716	109.63	0.2017	3.086	109.57	0.1997
210	6.330	111.44	.2097	4.734	111.38	.2066	3.774	111.32	.2042	3.135	111.26	.2022
220	6.427	113.13	.2122	4.807	113.07	.2091	3.832	113.01	.2067	3.184	112.95	.2047
230	6.525	114.83	.2147	4.880	114.77	.2116	3.891	114.71	.2092	3.234	114.65	.2072
240	6.622	116.54	.2172	4.952	116.48	.2141	3.950	116.42	.2117	3.283	116.36	.2097
250	6.719	118.26	0.2197	5.024	118.20	0.2166	4.010	118.14	0.2141	3.332	118.08	0.2121
260	6.816	119.99	.2221	5.096	119.93	.2190	4.068	119.87	.2166	3.382	119.81	.2146
270	6.913	121.73	.2245	5.168	121.67	.2214	4.126	121.61	.2190	3.431	121.55	.2170
280	7.011	123.48	.2269	5.240	123.42	.2238	4.185	123.37	.2214	3.479	123.31	.2194
290	7.108	125.25	.2293	5.313	125.19	.2261	4.243	125.15	.2237	3.528	125.09	.2218

Temp F	Pressure 20 psia Gage pressure 5.3 psig (Sat temp 134.6 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 159.1 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 177.9 F)			Pressure 50 psia Gage pressure 35.3 psig (Sat temp 193.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(1.625)*	*(98.56)*	*(0.1769)*	*(1.109)*	*(102.15)*	*(0.1787)*	*(0.8435)*	*(104.90)*	*(0.1802)*	*(0.6809)*	*(107.14)*	*(0.1816)*
200	1.827	109.32	0.1940	1.194	108.97	0.1894	0.8791	108.62	0.1860	0.6899	108.27	0.1833
210	1.857	111.01	.1966	1.215	110.67	.1919	.8953	110.32	.1885	.7031	109.97	.1858
220	1.887	112.70	.1991	1.236	112.37	.1945	.9114	112.03	.1911	.7163	111.69	.1884
230	1.917	114.40	.2016	1.257	114.08	.1970	.9274	113.74	.1936	.7296	113.41	.1909
240	1.947	116.12	.2041	1.278	115.80	.1995	.9435	115.47	.1951	.7430	115.14	.1934
250	1.977	117.84	0.2065	1.299	117.53	0.2019	0.9596	117.21	0.1986	0.7562	116.90	0.1959
260	2.007	119.58	.2090	1.320	119.28	.2044	.9756	118.97	.2010	.7693	118.67	.1984
270	2.037	121.33	.2114	1.341	121.03	.2068	0.9915	120.74	.2035	.7821	120.45	.2008
280	2.068	123.09	.2138	1.362	122.80	.2092	1.008	122.52	.2059	.7950	122.23	.2033
290	2.098	124.87	.2162	1.382	124.59	.2116	1.023	124.31	.2083	.8079	124.03	.2057
300	2.128	126.67	0.2185	1.402	126.39	0.2140	1.038	126.12	0.2107	0.8207	125.84	0.2081
310	2.158	128.47	.2209	1.422	128.20	.2163	1.054	127.94	.2131	.8335	127.66	.2104
320	2.187	130.28	.2232	1.442	130.02	.2187	1.070	129.76	.2154	.8463	129.48	.2128
330	2.217	132.10	.2255	1.462	131.85	.2210	1.085	131.59	.2178	.8591	131.31	.2152
340	2.247	133.94	.2278	1.483	133.69	.2233	1.101	133.44	.2201	.8716	133.16	.2175
350	2.277	135.78	0.2301	1.503	135.54	0.2256	1.117	135.29	0.2224	0.8844	135.02	0.2198
360	2.308	137.64	.2324	1.524	137.40	.2279	1.133	137.16	.2247	.8973	136.89	.2221
370	2.338	139.50	.2347	1.545	139.27	.2302	1.148	139.03	.2269	.9101	138.77	.2244
380	2.368	141.38	.2369	1.565	141.15	.2324	1.164	140.91	.2292	.9228	140.67	.2266
390	2.398	143.27	.2392	1.585	143.05	.2347	1.179	142.81	.2314	.9354	142.57	.2289


"FREON-114"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR.
VOLUME, CU FT PER LB
ENTROPY, BTU PER LB PER DEG. F
ENTHALPY - BTU PER LB, ABOVE -40 F
PRESSURE, PSIA
CONSTANT VOLUME
CONSTANT ENTROPY
SATURATED VAPOR
SATURATED LIQUID
KINETIC CHEMICALS INC.
0
10
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30
40
50
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65
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7.5
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4
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2
400°
360°
320°
280°
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160°
120°
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.75
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.17

Chart 12

Table 24. F-114 (CClF$_2$-CClF$_2$) Dichlorotetrafluoroethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−80	0.464	28.97*	105.603	51.26	−8.73	60.44	−0.0227	0.1595
−78	.50	28.90*	105.398	49.83	−8.30	60.71	− .0215	.1593
−76	.535	28.85*	105.193	44.87	−7.87	60.98	− .0203	.1592
−74	.58	28.73*	105.058	41.69	−7.44	61.25	− .0191	.1590
−72	.62	28.66*	104.924	38.92	−7.01	61.52	− .0179	.1589
−70	0.670	28.59*	104.790	36.40	−6.57	61.79	−0.0167	0.1587
−68	.72	28.46*	104.624	34.20	−6.14	62.06	− .0155	.1586
−66	.775	28.33*	104.458	31.74	−5.71	62.33	− .0143	.1585
−64	.833	28.23*	104.292	29.77	−5.28	62.60	− .0132	.1583
−62	.895	28.10*	104.126	27.79	−4.84	62.88	− .0121	.1582
−60	0.959	27.99*	103.960	26.06	−4.40	63.16	−0.0110	0.1580
−58	1.028	27.83*	103.792	24.55	−3.96	63.43	− .0098	.1579
−56	1.10	27.68*	103.622	22.94	−3.52	63.70	− .0087	.1578
−54	1.175	27.52*	103.452	21.50	−3.08	63.97	− .0075	.1577
−52	1.26	27.35*	103.282	20.16	−2.64	64.25	− .0064	.1576
−50	1.349	27.20*	103.113	18.96	−2.20	64.53	−0.0054	0.1575
−48	1.438	27.0*	102.938	17.85	−1.76	64.80	− .0043	.1574
−46	1.535	26.8*	102.766	16.80	−1.32	65.07	− .0032	.1573
−44	1.635	26.6*	102.594	15.75	−0.88	65.35	− .0021	.1572
−42	1.745	26.4*	102.422	14.87	−0.44	65.63	− .0010	.1571
−40	1.866	26.12*	102.25	14.02	0.00	65.91	0.0000	0.1571
−38	1.990	25.87*	102.08	13.20	0.45	66.19	.0011	.1570
−36	2.121	25.60*	101.90	12.44	0.91	66.47	.0021	.1569
−34	2.259	25.32*	101.72	11.73	1.36	66.74	.0032	.1568
−32	2.404	25.03*	101.55	11.07	1.81	67.02	.0042	.1567
−30	2.557	24.72*	101.37	10.45	2.27	67.30	0.0053	0.1567
−28	2.718	24.39*	101.19	9.877	2.72	67.58	.0063	.1567
−26	2.887	24.04*	101.01	9.338	3.17	67.86	.0074	.1566
−24	3.064	23.68*	100.83	8.833	3.63	68.14	.0084	.1565
−22	3.249	23.31*	100.65	8.362	4.08	68.42	.0095	.1565
−20	3.444	22.91*	100.47	7.921	4.54	68.70	0.0105	0.1565
−18	3.648	22.49*	100.29	7.508	4.99	68.98	.0116	.1565
−16	3.862	22.06*	100.11	7.121	5.44	69.26	.0126	.1564
−14	4.085	21.61*	99.92	6.757	5.90	69.54	.0136	.1564
−12	4.319	21.13*	99.74	6.416	6.35	69.82	.0146	.1564
−10	4.564	20.63*	99.56	6.095	6.81	70.10	0.0157	0.1564
− 8	4.819	20.11*	99.37	5.794	7.26	70.38	.0167	.1564
− 6	5.086	19.57*	99.19	5.510	7.72	70.66	.0177	.1564
− 4	5.365	19.00*	99.00	5.244	8.18	70.94	.0187	.1564
− 2	5.655	18.41*	98.81	4.992	8.63	71.22	.0197	.1565
0	5.958	17.79*	98.62	4.756	9.09	71.50	0.0207	0.1565
2	6.274	17.15*	98.44	4.533	9.54	71.78	.0217	.1565
4	6.603	16.48*	98.25	4.322	10.00	72.07	.0227	.1565
5†	6.772	16.14*	98.15	4.221	10.23	72.21	.0232	.1565
6	6.945	15.78*	98.06	4.123	10.46	72.35	.0236	.1566
8	7.301	15.06*	97.87	3.935	10.91	72.63	.0246	.1566
10	7.671	14.31*	97.68	3.758	11.37	72.91	0.0256	0.1566
12	8.057	13.52*	97.48	3.591	11.83	73.19	.0266	.1567
14	8.457	12.71*	97.29	3.432	12.29	73.47	.0275	.1567
16	8.873	11.86*	97.10	3.282	12.75	73.75	.0285	.1568
18	9.305	10.98*	96.90	3.140	13.20	74.04	.0295	.1568
20	9.753	10.07*	96.71	3.005	13.66	74.32	0.0304	0.1569
22	10.22	9.12*	96.51	2.877	14.12	74.60	.0314	.1569
24	10.70	8.14*	96.32	2.756	14.58	74.88	.0323	.1570
26	11.20	7.12*	96.12	2.641	15.05	75.17	.0333	.1571
28	11.72	6.07*	95.92	2.532	15.51	75.45	.0342	.1571

* Inches of mercury below one atmosphere.

† Standard ton temperatures.

Table 24. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Liquid and Saturated Vapor (Concluded)

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
30	12.25	4.99*	95.73	2.429	15.97	75.73	0.0352	0.1572
32	12.81	3.85*	95.53	2.330	16.43	76.01	.0361	.1573
34	13.38	2.69*	95.33	2.236	16.89	76.29	.0370	.1574
36	13.98	1.47*	95.13	2.147	17.36	76.58	.0380	.1575
38	14.59	0.22*	94.93	2.062	17.82	76.86	.0389	.1575
40	15.22	0.52	94.73	1.982	18.28	77.14	0.0398	0.1576
42	15.88	1.18	94.52	1.905	18.75	77.42	.0408	.1577
44	16.56	1.86	94.32	1.832	19.21	77.70	.0417	.1578
46	17.26	2.56	94.12	1.762	19.68	77.99	.0426	.1579
48	17.98	3.28	93.91	1.695	20.14	78.27	.0435	.1580
50	18.73	4.03	93.71	1.632	20.61	78.55	0.0444	0.1581
52	19.50	4.80	93.50	1.571	21.08	78.83	.0453	.1582
54	20.29	5.59	93.30	1.513	21.54	79.11	.0463	.1583
56	21.11	6.41	93.09	1.458	22.01	79.39	.0472	.1584
58	21.96	7.26	92.88	1.405	22.48	79.67	.0481	.1585
60	22.83	8.13	92.68	1.354	22.95	79.95	0.0490	0.1587
62	23.72	9.02	92.47	1.306	23.42	80.23	.0499	.1588
64	24.64	9.94	92.26	1.260	23.89	80.51	.0508	.1589
66	25.59	10.89	92.05	1.216	24.36	80.79	.0517	.1590
68	26.57	11.87	91.84	1.174	24.83	81.07	.0526	.1591
70	27.57	12.87	91.63	1.133	25.30	81.35	0.0534	0.1593
72	28.61	13.91	91.41	1.094	25.78	81.62	.0543	.1594
74	29.67	14.97	91.20	1.057	26.25	81.90	.0552	.1595
76	30.76	16.06	90.99	1.021	26.73	82.18	.0561	.1596
78	31.88	17.18	90.77	0.9869	27.20	82.46	.0570	.1597
80	33.04	18.34	90.56	0.9541	27.68	82.73	0.0579	0.1599
82	34.22	19.52	90.34	.9226	28.15	83.01	.0587	.1600
84	35.44	20.74	90.13	.8923	28.63	83.29	.0596	.1601
86†	36.69	21.99	89.91	.8632	29.11	83.56	.0605	.1603
88	37.97	23.27	89.69	.8353	29.58	83.84	.0613	.1604
90	39.29	24.59	89.47	0.8084	30.06	84.11	0.0622	0.1605
92	40.64	25.94	89.25	.7827	30.54	84.39	.0631	.1607
94	42.02	27.32	89.03	.7579	31.02	84.66	.0639	.1608
96	43.44	28.74	88.81	.7340	31.50	84.93	.0648	.1609
98	44.89	30.19	88.59	.7111	31.99	85.21	.0656	.1611
100	46.39	31.69	88.37	0.6890	32.47	85.48	0.0665	0.1612
102	47.92	33.22	88.15	.6677	32.95	85.75	.0674	.1614
104	49.48	34.78	87.93	.6472	33.43	86.02	.0682	.1615
106	51.09	36.39	87.70	.6274	33.92	86.29	.0691	.1617
108	52.73	38.03	87.48	.6084	34.40	86.56	.0699	.1618
110	54.41	39.71	87.25	0.5901	34.89	86.83	0.0708	0.1619
112	56.14	41.44	87.01	.5724	35.38	87.09	.0716	.1621
114	57.90	43.20	86.77	.5554	35.87	87.36	.0725	.1622
116	59.70	45.00	86.54	.5389	36.35	87.63	.0733	.1624
118	61.55	46.85	86.31	.5230	36.84	87.89	.0741	.1625
120	63.44	48.74	86.08	0.5077	37.33	88.16	0.0750	0.1627
122	65.37	50.67	85.85	.4929	37.83	88.42	.0758	.1628
124	67.35	52.65	85.61	.4787	38.32	88.69	.0767	.1630
126	69.37	54.67	85.37	.4649	38.81	88.95	.0775	.1631
128	71.43	56.73	85.13	.4515	39.31	89.21	.0783	.1633
130	73.54	58.84	84.89	0.4387	39.80	89.47	0.0792	0.1634
132	75.69	60.99	84.65	.4262	40.30	89.73	.0800	.1635
134	77.90	63.20	84.41	.4142	40.80	89.99	.0808	.1637
136	80.15	65.45	84.16	.4025	41.29	90.25	.0816	.1638
138	82.44	67.74	83.91	.3912	41.79	90.51	.0825	.1640
140	84.79	70.09	83.66	0.3803	42.29	90.76	0.0833	0.1641

* Inches of mercury below one atmosphere.
† Standard ton temperatures.

Table 25. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Superheated Vapor

Temp F	Pressure 1.5 psia Gage pressure 26.87 in. vac (Sat temp −46.6 F)			Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −37.8 F)			Pressure 2.5 psia Gage pressure 24.83 in. vac (Sat temp −30.7 F)			Pressure 3.0 psia Gage pressure 23.81 in. vac (Sat temp −24.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(17.18)*	*(65.00)*	*(0.1575)*	*(13.14)*	*(66.21)*	*(0.1570)*	*(10.68)*	*(67.20)*	*(0.1567)*	*(9.011)*	*(68.04)*	*(0.1566)*
−40	17.46	65.93	0.1596	—	—	—	—	—	—	—	—	—
−30	17.89	67.35	.1630	13.39	67.33	0.1596	10.70	67.30	0.1570	—	—	—
−20	18.31	68.78	.1663	13.71	68.76	.1629	10.95	68.74	.1603	9.108	68.72	0.1581
−10	18.73	70.23	.1695	14.03	70.21	.1662	11.20	70.18	.1635	9.230	70.16	.1614
0	19.15	71.69	0.1727	14.34	71.67	0.1694	11.46	71.64	0.1667	9.532	71.62	0.1646
10	19.57	73.16	.1759	14.66	73.14	.1725	11.71	73.12	.1699	9.744	73.10	.1678
20	19.99	74.65	.1790	14.98	74.63	.1757	11.96	74.61	.1741	9.956	74.59	.1709
30	20.42	76.15	.1821	15.29	76.13	.1788	12.22	76.11	.1762	10.17	76.09	.1740
40	20.84	77.66	.1852	15.61	77.65	.1818	12.47	77.63	.1792	10.38	77.61	.1771
50	21.26	79.19	0.1882	15.92	79.18	0.1849	12.73	79.16	0.1823	10.59	79.14	0.1801
60	21.68	80.73	.1912	16.24	80.72	.1879	12.98	80.70	.1853	10.80	80.68	.1831
70	22.10	82.29	.1942	16.56	82.28	.1908	13.23	82.26	.1882	11.01	82.24	.1861
80	22.52	83.86	.1971	16.87	83.85	.1938	13.49	83.83	.1912	11.23	83.81	.1890
90	22.94	85.45	.2000	17.19	85.43	.1967	13.74	85.42	.1941	11.44	85.40	.1919
100	23.36	87.05	0.2029	17.51	87.03	0.1996	13.99	87.02	0.1970	11.65	87.00	0.1948
110	23.78	88.66	.2058	17.82	88.64	.2024	14.25	88.63	.1998	11.86	88.61	.1977
120	24.20	90.29	.2086	18.14	90.27	.2053	14.50	90.26	.2026	12.07	90.24	.2005
130	24.62	91.93	.2114	18.45	91.91	.2081	14.75	91.90	.2054	12.28	91.88	.2033
140	25.04	93.58	.2142	18.77	93.57	.2108	15.00	93.55	.2082	12.49	93.54	.2061
150	25.46	95.25	0.2170	19.08	95.23	0.2136	15.26	95.22	0.2110	12.70	95.21	0.2089
160	25.88	96.93	.2197	19.40	96.91	.2163	15.51	96.90	.2137	12.92	96.89	.2116
170	26.30	98.62	.2224	19.71	98.61	.2190	15.76	98.60	.2164	13.13	98.58	.2143
180	26.72	100.33	.2251	20.03	100.32	.2217	16.01	100.31	.2191	13.34	100.29	.2170
190	27.14	102.06	.2278	20.35	102.04	.2244	16.27	102.03	.2218	13.55	102.02	.2197
200	27.56	103.79	0.2304	20.66	103.78	0.2271	16.52	103.77	0.2245	13.76	103.76	0.2223
210	27.98	105.54	.2331	20.98	105.53	.2297	16.77	105.52	.2271	13.97	105.51	.2250
220	28.40	107.31	.2357	21.29	107.30	.2323	17.02	107.28	.2297	14.18	107.27	.2276
230	28.82	109.09	.2383	21.61	109.08	.2349	17.28	109.06	.2323	14.39	109.05	.2302
240	29.24	110.88	.2409	21.92	110.87	.2375	17.53	110.86	.2349	14.60	110.84	.2328
250	29.66	112.09	0.2434	22.24	112.67	0.2401	17.78	112.66	0.2375	14.81	112.65	0.2353
260				22.55	114.49	0.2426	18.03	114.48	0.2400	15.02	114.67	.2379
270										15.23	116.31	0.2404

Temp F	Pressure 3.5 psia Gage pressure 22.80 in. vac (Sat temp −19.5 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp −14.8 F)			Pressure 4.5 psia Gage pressure 20.76 in. vac (Sat temp −10.5 F)			Pressure 5.0 psia Gage pressure 19.74 in. vac (Sat temp −6.6 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(7.806)*	*(66.77)*	*(0.1565)*	*(6.893)*	*(69.43)*	*(0.1565)*	*(6.177)*	*(70.02)*	*(0.1564)*	*(5.601)*	*(70.67)*	*(0.1564)*
−10	7.976	70.14	0.1596	6.967	70.12	0.1580	6.183	70.10	0.1566	—	—	—
0	8.158	71.60	0.1628	7.127	71.58	0.1612	6.326	71.56	0.1598	5.684	71.54	0.1585
10	8.340	73.08	.1660	7.287	73.06	.1644	6.468	73.04	.1630	5.813	73.02	.1617
20	8.522	74.57	.1691	7.446	74.55	.1675	6.610	74.53	.1661	5.941	74.51	.1649
30	8.704	76.07	.1722	7.606	76.05	.1706	6.752	76.04	.1692	6.069	76.02	.1680
40	8.886	77.59	.1753	7.765	77.57	.1737	6.894	77.56	.1723	6.197	77.54	.1711
50	9.068	79.12	0.1783	7.925	79.10	0.1767	7.036	79.09	0.1753	6.325	79.07	0.1741
60	9.250	80.67	.1813	8.085	80.65	.1797	7.178	80.63	.1783	6.453	80.61	.1771
70	9.431	82.23	.1843	8.244	82.21	.1827	7.320	82.19	.1813	6.581	82.17	.1801
80	9.613	83.80	.1872	8.403	83.78	.1856	7.462	83.76	.1843	6.709	83.75	.1830
90	9.795	85.38	.1901	8.562	85.37	.1885	7.604	85.35	.1872	6.837	85.33	.1859
100	9.976	86.98	0.1930	8.721	86.97	0.1914	7.745	86.95	0.1901	6.964	86.93	0.1888
110	10.16	88.60	.1959	8.880	88.58	.1943	7.887	88.57	.1929	7.092	88.55	.1917
120	10.34	90.23	.1987	9.039	90.21	.1971	8.028	90.20	.1957	7.219	90.19	.1945
130	10.52	91.87	.2015	9.198	91.85	.1999	8.170	91.84	.1985	7.346	91.82	.1973
140	10.70	93.52	.2043	9.356	93.51	.2027	8.311	93.49	.2013	7.474	93.48	.2001
150	10.88	95.19	0.2071	9.515	95.18	0.2055	8.452	95.16	0.2041	7.601	95.15	0.2029
160	11.06	96.87	.2098	9.673	96.86	.2082	8.593	96.84	.2068	7.728	96.83	.2056
170	11.24	98.57	.2125	9.832	98.56	.2109	8.734	98.54	.2096	7.855	98.53	.2083
180	11.43	100.28	.2152	9.990	100.27	.2136	8.875	100.25	.2123	7.983	100.24	.2110
190	11.61	102.00	.2179	10.15	101.99	.2163	9.015	101.98	.2149	8.110	101.96	.2137
200	11.79	103.74	0.2205	10.31	103.73	0.2190	9.156	103.72	0.2176	8.236	103.70	0.2163
210	11.97	105.49	.2232	10.47	105.48	.2216	9.297	105.47	.2202	8.363	105.46	.2190
220	12.15	107.26	.2258	10.62	107.25	.2242	9.438	107.24	.2228	8.490	107.22	.2216
230	12.33	109.04	.2284	10.78	109.03	.2268	9.579	109.02	.2254	8.617	109.00	.2242
240	12.51	110.83	.2310	10.94	110.82	.2294	9.720	110.81	.2280	8.744	110.80	.2268
250	12.69	112.64	0.2335	11.10	112.63	0.2320	9.861	112.62	0.2306	8.871	112.61	0.2294
260	12.87	114.46	.2361	11.26	114.45	.2345	10.00	114.44	.2331	8.998	114.43	.2319
270	13.05	116.30	.2386	11.42	116.28	.2371	10.14	116.27	.2357	9.125	116.26	.2344
280	13.23	118.15	0.2411	11.57	118.13	0.2396	10.28	118.12	0.2382	9.251	118.11	.2370
290										9.378	119.98	0.2395

Table 25. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane
Properties of Superheated Vapor (Continued)

Temp F	Pressure 5.5 psia Gage pressure 18.73 in. vac (Sat temp −3.1 F)			Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp +0.3 F)			Presure 6.5 psia Gage pressure 16.69 in. vac (Sat temp 3.4 F)			Pressure 7.0 psia Gage pressure 15.67 in. vac (Sat temp 6.3 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(5.126)*	*(71.07)*	*(0.1564)*	*(4.726)*	*(71.54)*	*(0.1565)*	*(4.386)*	*(71.98)*	*(0.1565)*	*(4.094)*	*(72.39)*	*(0.1566)*
0	5.159	71.52	0.1574									
10	5.276	73.00	.1606	4.829	72.98	0.1595	4.451	72.96	0.1586	4.127	72.94	0.1577
20	5.393	74.49	.1637	4.937	74.47	.1627	4.551	74.45	.1618	4.220	74.43	.1609
30	5.510	76.00	.1668	5.044	75.98	.1658	4.650	75.96	.1649	4.312	75.94	.1640
40	5.626	77.52	.1699	5.151	77.50	.1689	4.749	77.48	.1679	4.404	77.46	.1671
50	5.743	79.05	0.1730	5.258	79.03	0.1719	4.848	79.01	.1710	4.496	78.99	0.1701
60	5.860	80.60	.1760	5.365	80.58	.1749	4.947	80.56	.1740	4.588	80.54	.1731
70	5.976	82.16	.1789	5.472	82.14	.1779	5.046	82.12	.1770	4.680	82.10	.1761
80	6.092	83.73	.1819	5.579	83.71	.1809	5.145	83.70	.1799	4.772	83.69	.1790
90	6.209	85.32	.1848	5.686	85.30	.1838	5.244	85.28	.1828	4.864	85.27	.1819
100	6.325	86.92	0.1877	5.793	86.90	0.1867	5.342	86.89	0.1857	4.956	86.87	0.1848
110	6.441	88.53	.1905	5.899	88.52	.1895	5.441	88.50	.1886	5.048	88.49	.1877
120	6.557	90.16	.1934	6.006	90.15	.1924	5.539	90.13	.1914	5.139	90.12	.1905
130	6.673	91.81	.1962	6.112	91.79	.1952	5.638	91.78	.1942	5.231	91.76	.1933
140	6.789	93.46	.1990	6.219	93.45	.1980	5.736	93.43	.1970	5.322	93.42	.1961
150	6.905	95.13	0.2017	6.235	95.12	0.2007	5.834	95.10	0.1998	5.414	95.09	0.1989
160	7.021	96.82	.2045	6.431	96.80	.2035	5.932	96.79	.2025	5.505	96.77	.2016
170	7.136	98.52	.2072	6.537	98.50	.2062	6.031	98.49	.2052	5.596	98.47	.2043
180	7.252	100.23	.2099	6.644	100.21	.2089	6.129	100.29	.2079	5.687	100.18	.2070
190	7.368	101.95	.2126	6.750	101.94	.2115	6.227	101.92	.2106	5.778	101.91	.2097
200	7.483	103.69	9.2152	6.856	103.68	0.2142	6.325	103.66	0.2133	5.870	103.65	0.2124
210	7.599	105.44	.2179	6.962	105.43	.2168	6.423	105.42	.2159	5.961	105.40	.2150
220	7.714	107.21	.2205	7.068	107.20	.2195	6.521	107.19	.2185	6.052	107.17	.2176
230	7.830	108.99	.2231	7.173	108.98	.2221	6.619	108.97	.2211	6.143	108.95	.2202
240	7.945	110.78	.2257	7.279	110.77	.2246	6.716	110.76	.2237	6.234	110.75	.2228
250	8.061	112.59	0.2282	7.385	112.58	0.2272	6.814	112.57	0.2263	6.325	112.56	0.2254
260	8.176	114.42	.2308	7.491	114.40	.2298	6.912	114.39	.2288	6.416	114.38	.2279
270	8.291	116.25	.2333	7.597	116.24	.2323	7.010	116.23	.2314	6.507	116.22	.2305
280	8.407	118.10	.2358	7.703	118.09	.2348	7.108	118.08	.2339	6.597	118.07	.2330
290	8.522	119.96	0.2383	7.809	119.95	.2373	7.205	119.94	.2364	6.688	119.93	.2355
300				7.914	121.83	0.2398	7.303	121.82	0.2389	6.779	121.81	0.2380

Temp F	Pressure 7.5 psia Gage pressure 14.65 in. vac (Sat temp 9.1 F)			Pressure 8.0 psia Gage pressure 13.64 in. vac (Sat temp 11.7 F)			Pressure 9 psia Gage pressure 11.60 in. vac (Sat temp 16.6 F)			Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp 21.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(3.839)*	*(72.78)*	*(0.1566)*	*(3.616)*	*(73.15)*	*(0.1567)*	*(3.240)*	*(73.84)*	*(0.1568)*	*(2.937)*	*(74.47)*	*(0.1569)*
10	3.846	72.92	0.1569									
20	3.933	74.41	.1600	3.682	74.39	0.1593	3.263	74.35	0.1578			
30	4.019	75.92	.1631	3.763	75.90	.1624	3.336	75.86	.1609	2.994	75.82	0.1597
40	4.105	77.44	.1662	3.844	77.42	.1655	3.408	77.38	.1640	3.060	77.35	.1628
50	4.191	78.98	0.1693	3.925	78.96	0.1685	3.480	78.92	0.1671	3.125	78.88	0.1658
60	4.277	80.53	.1723	4.006	80.51	.1715	3.553	80.47	.1701	3.190	80.43	.1688
70	4.363	82.09	.1753	4.086	82.07	.1745	3.625	82.03	.1731	3.255	82.00	.1718
80	4.449	83.66	.1782	4.167	83.64	.1774	3.697	83.61	.1760	3.320	83.58	.1748
90	4.535	85.25	.1811	4.248	85.23	.1803	3.769	85.20	.1789	3.385	85.17	.1777
100	4.621	86.85	0.1840	4.328	86.84	0.1832	3.840	86.80	0.1818	3.450	86.77	0.1806
110	4.707	88.47	.1869	4.409	88.46	.1861	3.912	88.42	.1847	3.515	88.39	.1834
120	4.793	90.10	.1897	4.489	90.09	.1889	3.984	90.05	.1875	3.579	90.02	.1863
130	4.878	91.75	.1925	4.570	91.73	.1917	4.055	91.70	.1903	3.644	91.67	.1891
140	4.964	93.40	.1953	4.650	93.39	.1945	4.127	93.36	.1931	3.708	93.33	.1919
150	5.049	95.07	0.1981	4.730	95.06	0.1973	4.198	95.03	0.1959	3.773	95.00	0.1947
160	5.134	96.76	.2008	4.810	96.75	.2000	4.270	96.72	.1987	3.837	96.69	.1974
170	5.220	98.46	.2035	4.890	98.45	.2028	4.341	98.42	.2014	3.902	98.39	.2001
180	5.305	100.17	.2062	4.970	100.16	.2055	4.412	100.13	.2041	3.966	100.10	.2028
190	5.390	101.90	.2089	5.050	101.88	.2081	4.483	101.86	.2067	4.030	101.83	.2055
200	5.475	103.64	0.2116	5.130	103.62	0.2108	4.555	103.60	0.2094	4.095	103.57	0.2082
210	5.560	105.39	.2142	5.210	105.38	.2134	4.626	105.35	.2121	4.159	105.33	.2108
220	5.645	107.16	.2168	5.290	107.15	.2161	4.697	107.12	.2147	4.223	107.10	.2134
230	5.730	108.94	.2194	5.370	108.93	.2187	4.768	108.91	.2173	4.287	108.88	.2160
240	5.815	110.74	.2220	5.449	110.73	.2213	4.839	110.70	.2199	4.351	110.68	.2186
250	5.900	112.55	0.2246	5.529	112.54	0.2238	4.910	112.51	0.2224	4.415	112.49	0.2212
260	5.985	114.37	.2271	5.609	114.36	.2264	4.981	114.33	.2250	4.479	114.31	.2237
270	6.070	116.21	.2297	5.689	116.19	.2289	5.052	116.17	.2275	4.543	116.15	.2263
280	6.155	118.06	.2322	5.768	118.05	.2314	5.123	118.02	.2300	4.607	118.00	.2288
290	6.240	119.92	.2347	5.848	119.91	.2339	5.194	119.89	.2325	4.671	119.87	.2313
300	6.325	121.80	0.2372	5.927	121.79	0.2364	5.265	121.77	0.2350	4.735	121.75	0.2338
310				6.007	123.68	0.2389	5.336	123.66	0.2375	4.799	123.64	.2363
320										4.862	125.55	0.2387

Table 25. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Superheated Vapor (Continued)

Temp F	Pressure 11 psia Gage pressure 7.53 in. vac (Sat temp 25.2 F)			Pressure 12 psia Gage pressure 5.50 in. vac (Sat temp 29.1 F)			Pressure 13 psia Gage pressure 3.46 in. vac (Sat temp 32.7 F)			Pressure 14 psia Gage pressure 1.42 in. vac (Sat temp 36.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*2.686*)	(*75.05*)	(*0.1670*)	(*2.477*)	(*75.60*)	(*0.1672*)	(*2.298*)	(*76.11*)	(*0.1673*)	(*2.144*)	(*76.59*)	(*0.1676*)
30	2.714	75.78	0.1585	2.481	75.74	0.1575						
40	2.774	77.31	.1616	2.536	77.27	.1606	2.335	77.23	0.1596	2.162	77.19	0.1587
50	2.834	78.85	0.1647	2.591	78.81	0.1636	2.386	78.77	0.1626	2.210	78.73	0.1617
60	2.893	80.40	.1677	2.646	80.36	.1666	2.436	80.32	.1656	2.257	80.29	.1647
70	2.952	81.96	.1707	2.700	81.93	.1696	2.487	81.89	.1686	2.304	81.86	.1677
80	3.012	83.54	.1736	2.755	83.51	.1726	2.538	83.47	.1716	2.351	83.44	.1707
90	3.071	85.13	.1765	2.809	85.10	.1755	2.588	85.07	.1745	2.399	85.03	.1736
100	3.130	86.74	0.1794	2.864	86.71	0.1784	2.639	86.67	0.1774	2.446	86.64	0.1765
110	3.189	88.36	.1823	2.918	88.33	.1813	2.689	88.29	.1803	2.493	88.26	.1794
120	3.248	89.99	.1851	2.972	89.96	.1841	2.739	89.93	.1831	2.539	89.90	.1822
130	3.307	91.64	.1879	3.027	91.61	.1869	2.789	91.58	.1859	2.586	91.55	.1850
140	3.366	93.30	.1907	3.081	93.27	.1897	2.839	93.24	.1887	2.633	93.21	.1878
150	3.425	94.97	0.1935	3.135	94.94	0.1925	2.889	94.91	0.1915	2.679	94.88	0.1906
160	3.483	96.66	.1963	3.189	96.63	.1952	2.939	96.60	.1943	2.726	96.57	.1934
170	3.542	98.36	.1990	3.243	98.33	.1979	2.989	98.31	.1970	2.772	98.28	.1961
180	3.601	100.08	.2017	3.297	100.05	.2006	3.039	100.02	.1997	2.818	99.99	.1988
190	3.659	101.81	.2044	3.351	101.78	.2033	3.089	101.75	.2024	2.865	101.72	.2015
200	3.718	103.55	0.2070	3.404	103.52	0.2060	3.139	103.19	0.2050	2.911	103.47	0.2041
210	3.776	105.30	.2097	3.458	105.28	.2086	3.188	105.25	.2077	2.957	105.23	.2068
220	3.835	107.07	.2123	3.512	107.05	.2113	3.238	107.02	.2103	3.004	107.00	.2094
230	3.893	108.86	.2149	3.565	108.83	.2139	3.288	108.81	.2129	3.050	108.78	.2120
240	3.952	110.65	.2175	3.619	110.63	.2164	3.337	110.61	.2155	3.096	110.58	.2146
250	4.010	112.46	0.2201	3.672	112.44	0.2190	3.387	112.42	0.2181	3.142	112.39	0.2172
260	4.068	114.29	.2226	3.726	114.27	.2216	3.436	114.24	.2206	3.188	114.22	.2197
270	4.127	116.13	.2251	3.780	116.11	.2241	3.486	116.08	.2232	3.234	116.06	.2223
280	4.185	117.98	.2277	3.833	117.96	.2266	3.535	117.93	.2257	3.280	117.91	.2248
290	4.243	119.85	.2302	3.887	119.82	.2291	3.585	119.80	.2282	3.326	119.78	.2273
300	4.301	121.73	0.2327	3.940	121.70	0.2316	3.634	121.68	0.2307	3.372	121.66	0.2298
310	4.359	123.62	.2351	3.993	123.60	.2341	3.683	123.58	.2332	3.418	123.56	.2323
320	4.417	125.53	0.2376	4.047	125.51	0.2366	3.733	125.49	.2356	3.464	125.47	.2347
330							3.782	127.41	0.2381	3.510	127.39	0.2372

Temp F	Pressure 16 psia Gage pressure 1.3 psig (Sat temp 42.3 F)			Pressure 18 psia Gage pressure 3.3 psig (Sat temp 48.0 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 53.3 F)			Pressure 22 psia Gage pressure 7.3 psig (Sat temp 58.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*1.892*)	(*77.47*)	(*0.1577*)	(*1.694*)	(*78.27*)	(*0.1680*)	(*1.634*)	(*79.00*)	(*0.1683*)	(*1.409*)	(*79.68*)	(*0.1686*)
50	1.924	78.66	0.1601	1.701	78.58	0.1586						
60	1.965	80.21	.1631	1.738	80.14	.1616	1.557	80.06	0.1603	1.408	79.98	0.1591
70	2.007	81.78	.1661	1.776	81.71	.1646	1.591	81.64	.1633	1.439	81.56	.1621
80	2.049	83.37	.1690	1.813	83.30	.1676	1.624	83.22	.1663	1.470	83.15	.1651
90	2.090	84.96	.1720	1.850	84.90	.1705	1.658	84.82	.1692	1.501	84.75	.1681
100	2.131	86.57	0.1749	1.887	86.51	0.1734	1.692	86.44	0.1721	1.532	86.37	0.1709
110	2.173	88.20	.1778	1.924	88.13	.1763	1.725	88.07	.1750	1.562	88.00	.1738
120	2.214	89.83	.1806	1.961	89.77	.1792	1.759	89.71	.1779	1.593	89.64	.1767
130	2.255	91.48	.1834	1.998	91.42	.1820	1.792	91.36	.1807	1.623	91.30	.1795
140	2.296	93.15	.1862	2.035	93.09	.1848	1.825	93.03	.1835	1.654	92.97	.1823
150	2.337	94.83	0.1890	2.071	94.77	0.1876	1.858	94.71	0.1863	1.684	94.65	0.1851
160	2.378	96.52	.1918	2.108	96.46	.1903	1.891	96.40	.1890	1.714	96.34	.1879
170	2.419	98.22	.1945	2.144	98.16	.1931	1.924	98.11	.1918	1.745	98.05	.1906
180	2.460	99.94	.1972	2.181	99.88	.1958	1.957	99.83	.1945	1.775	99.77	.1933
190	2.500	100.67	.1999	2.217	101.62	.1984	1.990	101.56	.1972	1.805	101.51	.1960
200	2.541	103.41	0.2025	2.253	103.36	0.2011	2.023	103.31	0.1998	1.835	103.26	0.1987
210	2.582	105.17	.2052	2.290	105.12	.2038	2.056	105.07	.2025	1.865	105.02	.2013
220	2.622	106.95	.2078	2.326	106.90	.2064	2.089	106.84	.2051	1.895	106.79	.2040
230	2.663	108.73	.2104	2.362	108.68	.2090	2.122	108.63	.2077	1.925	108.58	.2066
240	2.704	110.53	.2130	2.398	110.48	.2116	2.155	110.43	.2103	1.955	110.39	.2092
250	2.744	112.35	0.2156	2.435	112.30	0.2142	2.187	112.25	0.2129	1.985	112.20	0.2117
260	2.785	114.17	.2181	2.471	114.13	.2167	2.220	114.08	.2155	2.014	114.03	.2143
270	2.825	116.01	.2207	2.507	115.97	.2193	2.252	115.92	.2180	2.044	115.88	.2168
280	2.866	117.87	.2232	2.543	117.82	.2218	2.285	117.78	.2205	2.074	117.73	.2194
290	2.906	119.74	.2257	2.579	119.69	.2243	2.317	119.65	.2230	2.104	119.60	.2219
300	2.946	121.62	0.2282	2.615	121.57	0.2268	2.350	121.53	0.2255	2.133	121.49	0.2244
310	2.986	123.51	.2307	2.651	123.47	.2293	2.382	123.43	.2280	2.163	123.39	.2269
320	3.027	125.42	.2332	2.687	125.38	.2317	2.415	125.34	.2305	2.192	125.30	.2293
330	3.067	127.35	.2356	2.723	127.30	.2342	2.447	127.26	.2329	2.222	127.22	.2318
340	3.107	129.28	0.2380	2.759	129.24	0.2366	2.480	129.20	.2354	2.252	129.16	.2342
350							2.512	131.15	0.2378	2.281	131.11	0.2367

Table 25. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane
Properties of Superheated Vapor (Continued)

Temp F	Pressure 24 psia Gage pressure 9.3 psig (Sat temp 62.6 F)			Pressure 26 psia Gage pressure 11.3 psig (Sat temp 66.8 F)			Pressure 28 psia Gage pressure 13.3 psig (Sat temp 70.8 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 74.6 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*1.292*)	(*80.91*)	(*0.1688*)	(*1.198*)	(*80.90*)	(*0.1690*)	(*1.117*)	(*81.46*)	(*0.1693*)	(*1.046*)	(*81.98*)	(*0.1696*)
70	1.313	81.49	0.1610	1.206	81.41	0.1600						
80	1.341	83.08	.1640	1.233	83.00	.1630	1.139	82.93	0.1620	1.058	82.85	0.1611
90	1.370	84.68	.1670	1.259	84.61	.1659	1.164	84.54	.1650	1.082	84.47	.1641
100	1.398	86.30	0.1699	1.286	86.23	0.1688	1.189	86.16	0.1679	1.105	86.09	0.1670
110	1.427	87.93	.1727	1.312	87.86	.1717	1.213	87.79	.1708	1.128	87.73	.1699
120	1.455	89.58	.1756	1.338	89.51	.1764	1.238	89.44	.1737	1.151	89.38	.1728
130	1.483	91.23	.1784	1.364	91.17	.1774	1.262	91.10	.1765	1.174	91.04	.1756
140	1.511	92.90	.1812	1.390	92.84	.1802	1.287	92.78	.1793	1.197	92.71	.1784
150	1.539	94.58	0.1840	1.416	94.52	0.1830	1.311	94.46	0.1821	1.220	94.40	0.1812
160	1.567	96.28	.1868	1.442	96.22	.1858	1.335	96.16	.1849	1.242	96.10	.1840
170	1.595	97.99	.1895	1.468	97.93	.1885	1.359	97.88	.1876	1.265	97.82	.1867
180	1.623	99.71	.1922	1.494	99.66	.1912	1.383	99.60	.1903	1.288	99.54	.1895
190	1.651	101.45	.1949	1.520	101.40	.1939	1.407	101.34	.1930	1.310	101.28	.1922
200	1.678	103.20	0.1976	1.545	103.15	0.1966	1.431	103.09	0.1957	1.333	103.04	0.1949
210	1.706	104.96	.2003	1.571	104.91	.1993	1.455	104.86	.1984	1.355	104.81	.1975
220	1.733	106.74	.2029	1.596	106.69	.2019	1.479	106.64	.2010	1.378	106.59	.2002
230	1.761	108.53	.2055	1.622	108.48	.2045	1.503	108.43	.2036	1.400	108.38	.2028
240	1.788	110.34	.2081	1.647	110.29	.2071	1.527	110.24	.2062	1.422	110.19	.2054
250	1.816	112.15	0.2107	1.673	112.11	0.2097	1.550	112.06	0.2088	1.444	112.01	0.2079
260	1.843	113.98	.2133	1.698	113.94	.2123	1.574	113.89	.2114	1.467	113.84	.2105
270	1.871	115.83	.2158	1.724	115.78	.2148	1.598	115.74	.2139	1.489	115.69	.2131
280	1.898	117.69	.2183	1.749	117.64	.2174	1.622	117.60	.2164	1.511	117.55	.2156
290	1.925	119.56	.2208	1.774	119.51	.2199	1.645	119.47	.2190	1.533	119.42	.2181
300	1.953	121.44	0.2233	1.800	121.40	0.2224	1.669	121.36	0.2215	1.555	121.31	0.2206
310	1.980	123.34	.2258	1.825	123.30	.2249	1.692	123.26	.2240	1.577	123.21	.2231
320	2.007	125.25	.2283	1.850	125.21	.2273	1.716	125.17	.2264	1.599	125.13	.2256
330	2.034	127.18	.2307	1.875	127.14	.2298	1.739	127.10	.2289	1.621	127.06	.2280
340	2.061	129.12	.2332	1.900	129.08	.2322	1.762	129.04	.2313	1.643	129.00	.2305
350	2.088	131.07	0.2356	1.925	131.04	0.2346	1.786	131.04	0.2337	1.665	130.96	0.2329
360	2.116	133.04	0.2380	1.951	133.01	0.2371	1.809	132.97	.2362	1.687	132.93	.2353
370							1.833	134.95	0.2386	1.708	134.91	0.2377

Temp F	Pressure 32 psia Gage pressure 17.3 psig (Sat temp 78.2 F)			Pressure 34 psia Gage pressure 19.3 psig (Sat temp 81.6 F)			Pressure 36 psia Gage pressure 21.3 psig (Sat temp 84.9 F)			Pressure 38 psia Gage pressure 23.3 psig (Sat temp 88.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.9836*)	(*82.49*)	(*0.1698*)	(*0.9284*)	(*82.96*)	(*0.1600*)	(*0.8792*)	(*83.41*)	(*0.1602*)	(*0.8347*)	(*83.84*)	(*0.1604*)
80	0.9874	82.78	0.1603									
90	1.010	84.39	.1633	0.9458	84.32	0.1625	0.8891	84.24	0.1617	0.8384	84.17	0.1610
100	1.032	86.02	0.1662	0.9666	85.95	0.1654	0.9089	85.87	0.1646	0.8573	85.80	0.1639
110	1.053	87.66	.1691	0.9873	87.59	.1683	.9286	87.52	.1675	.8761	87.44	.1668
120	1.075	89.31	.1720	1.008	89.24	.1712	.9482	89.17	.1704	.8948	89.10	.1697
130	1.097	90.97	.1748	1.028	90.91	.1740	.9677	90.84	.1733	.9134	90.77	.1726
140	1.118	92.65	.1776	1.049	92.59	.1768	0.9871	92.52	.1761	.9320	92.46	.1754
150	1.140	94.34	0.1804	1.069	94.28	0.1796	1.007	94.21	0.1789	0.9505	94.15	0.1782
160	1.161	96.04	.1832	1.090	95.98	.1824	1.026	95.92	.1817	.9689	95.86	.1810
170	1.183	97.76	.1859	1.110	97.70	.1852	1.045	97.64	.1844	0.9872	97.58	.1837
180	1.204	99.49	.1887	1.130	99.43	.1879	1.064	99.37	.1872	1.005	99.31	.1865
190	1.225	101.23	.1914	1.150	101.17	.1906	1.083	101.11	.1899	1.024	101.06	.1892
200	1.246	102.98	0.1940	1.170	102.93	0.1933	1.102	102.87	0.1926	1.042	102.82	0.1919
210	1.268	104.75	.1967	1.190	104.70	.1959	1.121	104.64	.1952	1.060	104.59	.1945
220	1.289	106.53	.1993	1.210	106.48	.1986	1.140	106.43	.1979	1.078	106.38	.1972
230	1.310	108.33	.2020	1.230	108.28	.2012	1.159	108.23	.2005	1.096	108.18	.1998
240	1.331	110.14	.2046	1.250	110.09	.2038	1.178	110.04	.2031	1.113	109.99	.2024
250	1.352	111.96	0.2072	1.269	111.91	0.2064	1.197	111.86	0.2057	1.131	111.81	0.2050
260	1.372	113.79	.2097	1.289	113.75	.2090	1.215	113.70	.2083	1.149	113.65	.2076
270	1.393	115.64	.2123	1.309	115.60	.2115	1.234	115.55	.2108	1.167	115.50	.2101
280	1.414	117.50	.2148	1.328	117.46	.2141	1.253	117.41	.2133	1.185	117.37	.2127
290	1.435	119.38	.2173	1.348	119.33	.2166	1.271	119.29	.2159	1.203	119.25	.2152
300	1.455	121.27	0.2198	1.368	121.22	0.2191	1.290	121.18	0.2184	1.220	121.14	0.2177
310	1.476	123.17	.2223	1.387	123.13	.2216	1.308	123.08	.2209	1.238	123.04	.2202
320	1.497	125.09	.2248	1.407	125.05	.2241	1.327	125.00	.2233	1.255	124.96	.2227
330	1.517	127.02	.2272	1.426	126.98	.2265	1.345	126.93	.2258	1.273	126.89	.2251
340	1.538	128.96	.2297	1.446	128.92	.2289	1.364	128.88	.2283	1.290	128.84	.2276
350	1.559	130.92	0.2321	1.465	130.88	0.2314	1.382	130.84	0.2307	1.308	130.80	0.2300
360	1.579	132.89	.2345	1.485	132.85	.2338	1.401	132.81	.2331	1.325	132.77	.2324
370	1.600	134.87	0.2370	1.504	134.83	.2362	1.419	134.79	.2355	1.343	134.75	.2348
380				1.524	136.83	0.2386	1.437	136.79	0.2379	1.360	136.75	0.2372

Table 25. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Superheated Vapor (Continued)

emp F	Pressure 40 psia Gage pressure 25.3 psig (Sat temp 91.0 F)			Pressure 45 psia Gage pressure 30.3 psig (Sat temp 98.1 F)			Pressure 50 psia Gage pressure 35.3 psig (Sat temp 104.6 F)			Pressure 55 psia Gage pressure 40.3 psig (Sat temp 110.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
t sat)	(0.7948)	(84.25)	(0.1606)	(0.7096)	(85.23)	(0.6111)	(0.6408)	(86.10)	(0.1616)	(0.5841)	(86.92)	(0.1620)
100	0.8108	85.73	0.1632	0.7125	85.54	0.1616						
110	.8288	87.37	.1661	.7288	87.19	.1646	0.6487	87.00	0.1631			
120	.8467	89.03	.1690	.7450	88.86	.1675	.6636	88.68	.1660	0.5967	88.49	0.1647
130	.8646	90.70	.1719	.7611	90.53	.1703	.6784	90.36	.1689	.6104	90.18	.1676
140	.8824	92.39	.1747	.7772	92.22	.1732	.6931	92.05	.1718	.6240	91.88	.1705
150	0.9001	94.09	0.1775	0.7932	93.92	0.1760	0.7077	93.76	0.1746	0.6375	93.59	0.1733
160	.9176	95.80	.1803	.8091	95.64	.1788	.7223	95.48	.1774	.6509	95.31	.1761
170	.9351	97.52	.1831	.8250	97.37	.1816	.7367	97.21	.1802	.6643	97.05	.1789
180	.9524	99.25	.1858	.8407	99.11	.1843	.7511	98.95	.1829	.6776	98.80	.1816
190	.9697	101.00	.1885	.8563	100.86	.1870	.7654	100.71	.1856	.6908	100.56	.1844
200	0.9870	102.76	0.1912	0.8719	102.62	0.1897	0.7796	102.48	0.1883	0.7039	102.33	0.1871
210	1.004	104.53	.1939	.8874	104.04	.1924	.7937	104.26	.1910	.7170	104.12	.1898
220	1.021	106.32	.1965	.9028	106.19	.1950	.8077	106.05	.1937	.7300	105.91	.1924
230	1.039	108.12	.1992	.9182	107.99	.1977	.8217	107.86	.1963	.7429	107.72	.1951
240	1.056	109.93	.2018	.9335	109.81	.2003	.8357	109.68	.1989	.7558	109.55	.1977
250	1.073	111.76	0.2044	0.9488	111.64	0.2029	0.8496	111.51	0.2015	0.7686	111.38	0.2003
260	1.090	113.60	.2069	.9641	113.48	.2055	.8635	113.35	.2041	.7813	113.23	.2029
270	1.107	115.45	.2095	.9793	115.33	.2080	.8773	115.21	.2067	.7940	115.09	.2054
280	1.124	117.32	.2120	0.9945	117.20	.2106	.8911	117.08	.2092	.8066	116.97	.2080
290	1.141	119.20	.2146	1.010	119.08	.2131	.9049	118.97	.2118	.8192	118.85	.2105
300	1.157	121.09	0.2171	1.025	120.98	0.2156	0.9186	120.87	0.2143	0.8318	120.75	0.2130
310	1.174	123.00	.2196	1.040	122.89	.2181	.9323	122.78	.2168	.8444	122.67	.2155
320	1.191	124.92	.2220	1.055	124.81	.2206	.9460	124.70	.2193	.8569	124.59	.2180
330	1.208	126.85	.2245	1.070	126.75	.2230	.9597	126.64	.2217	.8694	126.53	.2205
340	1.224	128.80	.2270	1.085	128.70	.2255	.9733	128.59	.2242	.8819	128.49	.2230
350	1.241	130.76	0.2294	1.100	130.66	0.2279	0.9868	130.55	0.2266	0.8943	130.45	0.2254
360	1.258	132.73	.2318	1.115	132.63	.2304	1.000	132.53	.2290	.9067	132.43	.2278
370	1.274	134.72	.2342	1.130	134.62	.2328	1.014	134.52	.2315	.9191	134.42	.2303
380	1.291	136.72	.2366	1.144	136.62	.2352	1.027	136.53	.2339	.9314	136.43	.2327
390	1.308	138.73	0.2390	1.159	138.63	0.2376	1.041	138.54	.2362	.9437	138.45	.2351
400							1.054	140.57	0.2386	0.9560	140.48	0.2374
410										0.9683	142.53	0.2398

Temp F	Pressure 60 psia Gage pressure 45.3 psig (Sat temp 116.3 F)			Pressure 65 psia Gage pressure 50.3 psig (Sat temp 121.6 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 126.6 F)			Pressure 75 psia Gage pressure 60.3 psig (Sat temp 131.4 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
at sat)	(0.5364)	(87.67)	(0.1624)	(0.4958)	(88.37)	(0.1628)	(0.4607)	(89.03)	(0.1632)	(0.4302)	(89.65)	(0.1636)
120	0.5411	88.30	0.1635									
130	.5539	90.00	.1664	0.5059	89.81	0.1652	0.4645	89.62	0.1642			
140	.5666	91.70	.1693	.5179	91.52	.1681	.4759	91.34	.1670	0.4395	91.15	0.1660
150	0.5792	93.42	0.1721	0.5298	93.25	0.1710	0.4873	93.07	0.1699	0.4502	92.89	0.1689
160	.5918	95.15	.1749	.5415	94.98	.1738	.4985	94.81	.1727	.4608	94.63	.1717
170	.6043	96.89	.1777	.5532	96.73	.1766	.5095	96.56	.1755	.4713	96.39	.1745
180	.6166	98.64	.1805	.5648	98.49	.1794	.5204	98.33	.1783	.4817	98.16	.1773
190	.6288	100.41	.1832	.5763	100.26	.1821	.5312	100.10	.1811	.4920	99.94	.1801
200	0.6410	102.18	0.1859	0.5876	102.04	0.1848	0.5419	101.88	0.1838	0.5022	101.73	0.1828
210	.6530	103.97	.1886	.5989	103.83	.1875	.5525	103.68	.1865	.5122	103.53	.1855
220	.6650	105.78	.1913	.6101	105.63	.1902	.5630	105.49	.1892	.5222	105.35	.1882
230	.6770	107.59	.1939	.6213	107.45	.1928	.5735	107.31	.1918	.5322	107.17	.1909
240	.6889	109.41	.1965	.6324	109.28	.1955	.5840	109.15	.1945	.5421	109.01	.1935
250	0.7008	111.25	0.1991	0.6435	111.12	0.1981	0.5944	110.99	0.1971	0.5519	110.86	0.1962
260	.7126	113.10	.2017	.6545	112.98	.2007	.6048	112.85	.1997	.5617	112.72	.1988
270	.7244	114.97	.2043	.6655	114.85	.2033	.6151	114.72	.2023	.5715	114.59	.2014
280	.7362	116.85	.2069	.6765	116.73	.2058	.6254	116.60	.2048	.5812	116.48	.2040
290	.7479	118.74	.2094	.6874	118.62	.2084	.6356	118.50	.2074	.5908	118.38	.2065
300	0.7595	120.64	0.2119	0.6983	120.52	0.2109	0.6458	120.41	0.2099	0.6004	120.29	0.2090
310	.7711	122.56	.2144	.7091	122.44	.2134	.6559	122.33	.2124	.6099	122.22	.2115
320	.7827	124.49	.2169	.7199	124.37	.2159	.6660	124.26	.2149	.6194	124.15	.2140
330	.7943	126.43	.2194	.7307	126.32	.2184	.6761	126.21	.2174	.6289	126.10	.2165
340	.8058	128.38	.2219	.7414	128.28	.2208	.6862	128.17	.2199	.6383	128.06	.2190

Table 25. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane Properties of Superheated Vapor (Concluded)

Temp F	Pressure 60 psia Gage pressure 15.3 psig (Sat temp 116.3 F)			Pressure 65 psia Gage pressure 50.3 psig (Sat temp 121.6 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 126.6 F)			Pressure 75 psia Gage pressure 60.3 psig (Sat temp 131.4 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.5364*)	(*87.67*)	(*0.1624*)	(*0.4958*)	(*88.37*)	(*0.1628*)	(*0.4807*)	(*89.03*)	(*0.1632*)	(*0.4502*	(*89.66*)	(*0.1635*)
350	0.8173	130.35	0.2243	0.7521	130.25	0.2233	0.6962	130.14	0.2223	0.6477	130.04	0.2214
360	.8287	132.33	.2267	.7627	132.23	.2257	.7062	132.13	.2248	.6571	132.03	.2239
370	.8401	134.33	.2292	.7733	134.23	.2281	.7161	134.13	.2272	.6664	134.03	.2263
380	.8515	136.34	.2316	.7839	136.24	.2305	.7260	136.14	.2296	.6757	136.04	.2287
390	.8629	138.36	.2340	.7945	138.26	.2329	.7359	138.17	.2320	.6850	138.07	.2311
400	0.8742	140.39	0.2363	0.8050	140.30	0.2353	0.7458	140.21	0.2344	0.6943	140.11	0.2335
410	0.8855	142.44	0.2387	.8155	142.35	.2377	.7556	142.26	.2368	.7036	142.16	.2359
420				0.8260	144.41	0.2401	0.7654	144.32	0.2391	.7128	144.23	.2382
430										0.7220	146.31	0.2406

Temp F	Pressure 80 psia Gage pressure 65.3 psig (Sat temp 135.9 F)			Pressure 85 psia Gage pressure 70.3 psig (Sat temp 140.2 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.4033*)	(*90.24*)	(*0.1638*)	(*0.3794*)	(*90.78*)	(*0.1641*)
140	0.4076	90.96	0.1650	—	—	—
150	0.4178	92.70	0.1679	0.3890	92.51	0.1670
160	.4279	94.46	.1708	.3988	94.27	.1699
170	.4379	96.22	.1736	.4084	96.04	.1727
180	.4478	97.99	.1764	.4179	97.83	.1755
190	.4576	99.78	.1792	.4273	99.62	.1783
200	0.4674	101.58	0.1819	0.4366	101.42	0.1811
210	.4770	103.38	.1846	.4458	103.23	.1838
220	.4865	105.20	.1873	.4548	105.05	.1865
230	.4959	107.03	.1900	.4638	106.89	.1892
240	.5053	108.87	.1927	.4728	108.73	.1918
250	0.5146	110.72	0.1953	0.4817	110.59	0.1944
260	.5239	112.59	.1979	.4905	112.46	.1971
270	.5331	114.47	.2005	.4993	114.34	.1996
280	.5423	116.36	.2031	.5081	116.23	.2022
290	.5514	118.26	.2056	.5168	118.14	.2048
300	0.5605	120.17	0.2081	0.5254	120.05	0.2073
310	.5695	122.10	.2107	.5340	121.98	.2099
320	.5785	124.04	.2132	.5426	123.93	.2124
330	.5875	125.99	.2157	.5511	125.88	.2149
340	.5964	127.96	.2181	.5595	127.85	.2173
350	0.6053	129.93	0.2206	0.5679	129.83	0.2198
360	.6142	131.92	.2230	.5763	131.82	.2222
370	.6230	133.93	.2255	.5847	133.83	.2247
380	.6318	135.94	.2279	.5930	135.85	.2271
390	.6406	137.97	.2303	.6013	137.88	.2295
400	0.6494	140.02	0.2327	0.6096	139.92	0.2319
410	.6581	142.07	.2351	.6179	141.98	.2343
420	.6668	144.14	.2374	.6262	144.05	.2366
430	.6775	146.22	.2398	.6344	146.13	.2390
440	—	—	—	.6426	148.23	.2413

Table 26. Isobutane—Properties of Liquid and Saturated Vapor

Temp F	Pressure†		Liquid, density**	Vapor, sp vol**	Enthalpy, datum 0 F Btu per lb		Entropy, datum 0 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−20	7.50	14.6*	38.35	11.0	−9.0	156.5	−0.020	0.356
−15	8.30	13.0*	38.15	9.90	−7.0	157.0	− .015	.354
−10	9.28	11.0*	37.95	8.91	−4.5	158.5	− .010	.353
− 5	10.4	8.8*	37.80	7.99	−2.5	159.5	− .005	.351
0	11.6	6.3*	37.60	7.17	0.0	160.5	0.000	0.350
+ 5	13.1	3.3*	37.40	6.41	+2.5	162.0	0.005	0.348
10	14.6	0.2*	37.20	5.75	5.0	163.5	.011	.348
15	16.3	1.6	37.00	5.18	7.5	164.5	.016	.347
20	18.2	3.5	36.80	4.68	10.0	166.0	.021	.346
25	20.2	5.5	36.60	4.24	13.0	167.5	.027	.346
30	22.3	7.6	36.40	3.86	15.5	169.0	0.032	0.346
35	24.6	9.9	36.20	3.52	18.0	170.5	.038	.346
40	26.9	12.2	36.00	3.22	21.0	172.0	.044	.346
45	29.5	14.8	35.80	2.96	24.0	174.0	.049	.346
50	32.5	17.8	35.60	2.71	27.0	175.5	.055	.346
55	35.5	20.8	35.40	2.49	30.0	177.5	0.061	0.347
60	38.7	24.0	35.20	2.28	33.0	179.0	.067	.348
65	42.2	27.5	35.00	2.10	36.5	181.0	.073	.349
70	45.8	31.1	34.80	1.94	39.5	133.0	.079	.350
75	49.7	35.0	34.60	1.79	43.0	185.0	.086	.351
80	53.9	39.2	34.35	1.66	46.5	187.0	0.092	0.352
85	58.6	43.9	34.10	1.54	50.0	189.0	.098	.353
90	63.3	48.6	33.90	1.42	53.5	191.0	.105	.356
95	68.4	53.7	33.70	1.32	57.5	193.5	.112	.358
100	73.7	59.0	33.45	1.23	61.0	195.5	.118	.359
105	79.3	64.6	33.25	1.14	65.0	198.0	0.125	0.360
110	85.1	70.4	33.00	1.07	69.0	200.0	.132	.362
115	91.4	76.7	32.80	0.990	73.0	202.5	.139	.364
120	98.0	83.3	32.50	.926	77.0	204.5	.147	.367
125	104.8	90.1	32.30	.867	81.5	207.5	.154	.369
130	112.0	97.3	32.00	0.811	86.0	209.0	0.161	0.371
135	119.3	104.6	31.80	.760	90.5	211.0	.169	.375
140	126.8	112.1	31.50	.710	95.0	213.2	.176	.377
145	136.0	121.3	31.30	.662	99.6	215.2	.183	.379
150	145.0	130.3	31.03	.620	104.5	217.5	.190	.382
155	155.0	140.3	30.73	0.580	109.3	219.7	0.197	0.384
160	165.0	150.3	30.43	.542	114.2	221.7	.204	.387
165	175.0	160.3	30.14	.505	119.3	224.0	.212	.389
170	186.0	171.3	29.85	.475	124.7	225.7	.219	.392
175	198.0	183.3	29.54	.448	129.9	228.0	.226	.395
180	210.0	195.3	29.23	0.420	135.0	230.0	0.233	0.397

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† Based on material from Dana, Jenkins, Burdick and Timm, published originally in *Refrigerating Engineering*, June 1926, vol. 12, no. 12, page 402.

* Inches of mercury below one standard atmosphere (29.92 in.).

** Specific volume and density beyond 135 F, B. H. Sage and W. N. Lacey, *Industrial and Engineering Chemistry*, June 1938.

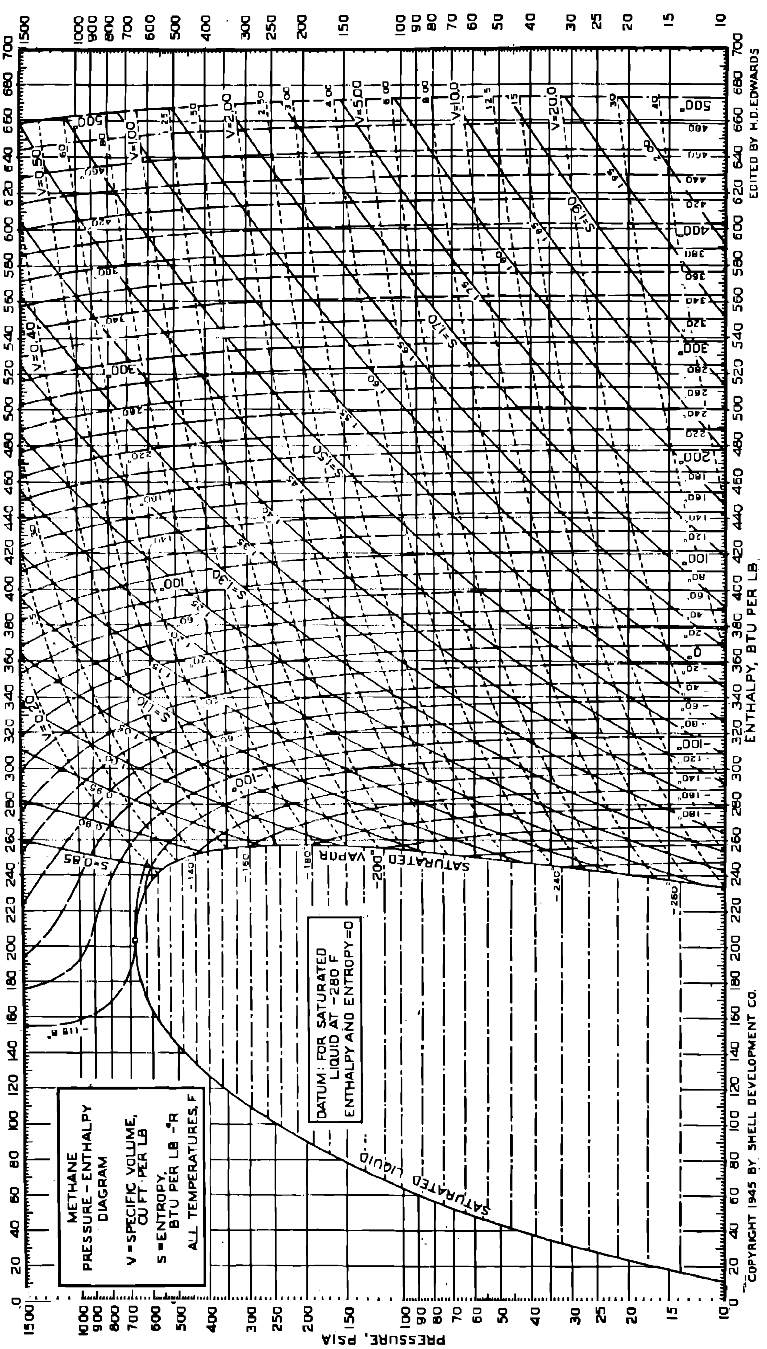
METHANE
PRESSURE - ENTHALPY
DIAGRAM
V = SPECIFIC VOLUME, CU FT PER LB
S = ENTROPY, BTU PER LB -°R
ALL TEMPERATURES, F
DATUM: FOR SATURATED LIQUID AT -280 F ENTHALPY AND ENTROPY = 0
SATURATED LIQUID
SATURATED VAPOR
ENTHALPY, BTU PER LB
PRESSURE, PSIA
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360 380 400 420 440 460 480 500 520 540 560 580 600 620 640 660 680 700
1500 1000 900 800 700 600 500 400 300 250 200 150 100 90 80 70 60 50 40 30 25 20 15 10
V=0.20
V=0.40
V=0.50
V=1.00
V=2.00
V=5.00
V=10.0
V=20.0
S=0.85
S=1.10
S=1.20
S=1.30
S=1.40
S=1.50
S=1.60
S=1.70
S=1.80
S=1.90
-280 -260 -240 -200 -180 -160 -140 -120 -100 -80 -60 -40 -20 0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 360 380 400 420 440 460 480 500
COPYRIGHT 1945 BY SHELL DEVELOPMENT CO.
EDITED BY H.D. EDWARDS

Chart 13

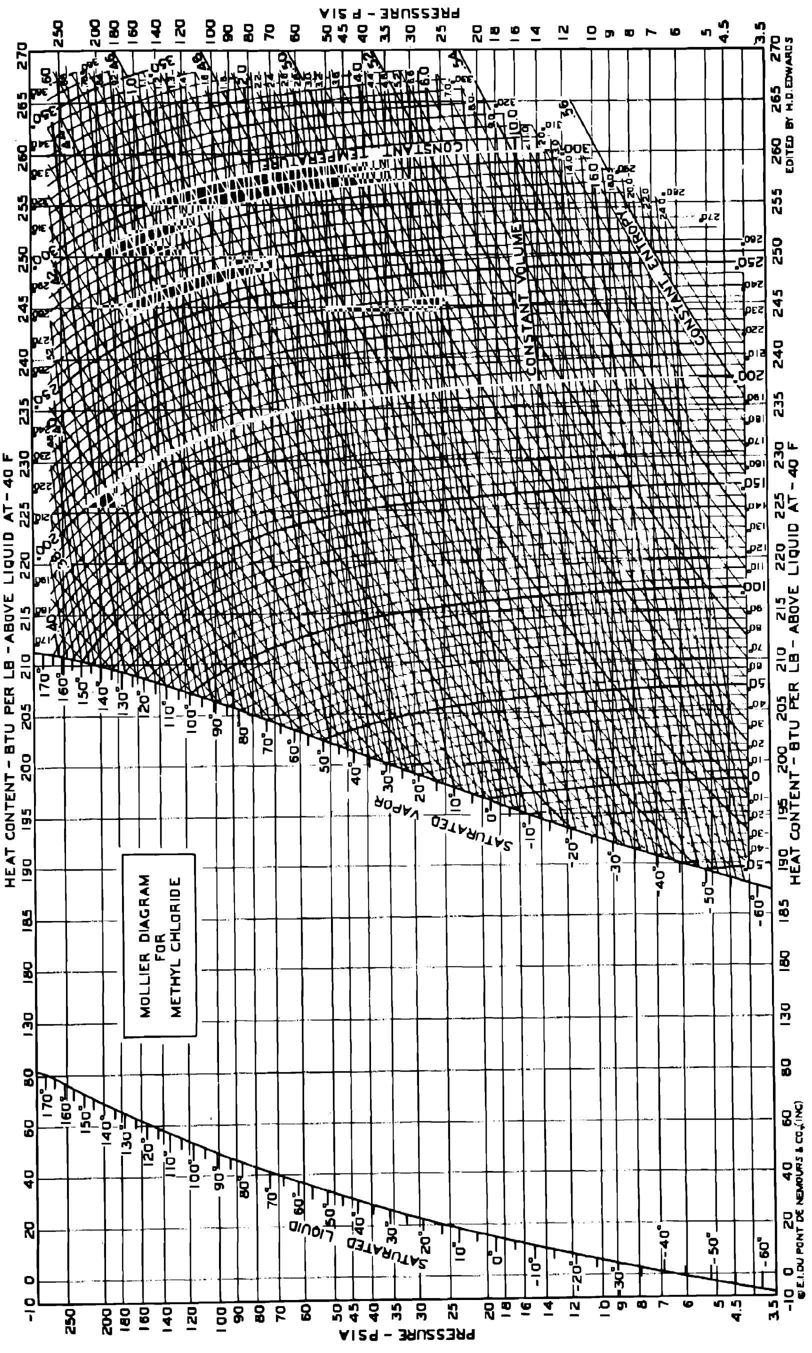
MOLLIER DIAGRAM FOR METHYL CHLORIDE
HEAT CONTENT - BTU PER LB - ABOVE LIQUID AT - 40 F
PRESSURE - PSIA
SATURATED LIQUID
SATURATED VAPOR
CONSTANT TEMPERATURE
CONSTANT VOLUME
CONSTANT ENTROPY
EDITED BY H.D. EDWARDS
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Chart 14

Table 27. Methyl Chloride—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−80	1.953	25.94*	0.01493	41.08	−13.888	184.75	−0.0351	0.4882
−70	2.715	24.32*	.01508	29.84	−10.521	186.25	− .0261	.4790
−60	3.799	22.19*	.01523	22.09	− 7.039	187.74	− .0172	.4703
−50	5.155	19.43*	.01538	16.64	− 3.532	189.19	− .0085	.4620
−40	6.878	15.92*	0.01553	12.72	0.000	190.66	0.0000	0.4544
−30	9.036	11.52*	.01568	9.873	3.562	192.08	.0084	.4472
−20	11.71	6.090*	.01583	7.761	7.146	193.49	.0166	.4405
−10	14.96	0.266	.01598	6.176	10.75	194.87	.0247	.4343
0	18.90	4.201	0.01613	4.969	14.39	196.23	0.0327	0.4284
2	19.77	5.077	.01616	4.763	15.12	196.51	.0343	.4273
4	20.68	5.985	.01619	4.568	15.85	196.78	.0359	.4262
5	21.15	6.455	.01622	4.471	16.21	196.92	.0367	.4257
6	21.62	6.924	.01625	4.379	16.58	197.05	.0375	.4251
8	22.59	7.896	.01628	4.206	17.31	197.31	.0390	.4240
10	23.60	8.903	0.01631	4.038	18.04	197.58	0.0406	0.4229
12	24.64	9.943	.01634	3.878	18.77	197.83	.0422	.4218
14	25.72	11.02	.01637	3.726	19.51	198.09	.0437	.4208
16	26.83	12.13	.01640	3.581	20.25	198.34	.0453	.4198
18	27.97	13.28	.01644	3.443	20.98	198.59	.0468	.4187
20	29.16	14.46	0.01647	3.312	21.73	198.84	0.0484	0.4177
22	30.38	15.69	.01650	3.186	22.47	199.08	.0499	.4166
24	31.64	16.95	.01654	3.067	23.21	199.32	.0514	.4156
26	32.95	18.25	.01658	2.952	23.95	199.56	.0530	.4146
28	34.29	19.60	.01662	2.843	24.70	199.79	.0545	.4136
30	35.68	20.98	0.01665	2.739	25.44	200.03	0.0560	0.4126
32	37.11	22.41	.01669	2.640	26.18	200.26	.0575	.4117
34	38.58	23.88	.01673	2.546	26.93	200.49	.0590	.4107
36	40.09	25.39	.01677	2.455	27.67	200.72	.0605	.4098
38	41.65	26.95	.01681	2.369	28.42	200.95	.0621	.4088
40	43.25	28.56	0.01684	2.286	29.17	201.17	0.0636	0.4079
42	44.91	30.21	.01688	2.206	29.92	201.40	.0651	.4070
44	46.61	31.91	.01692	2.130	30.67	201.62	.0665	.4061
46	48.35	33.66	.01696	2.057	31.42	201.84	.0680	.4052
48	50.15	35.45	.01700	1.987	32.17	202.06	.0695	.4043
50	51.99	37.29	0.01704	1.920	32.93	202.28	0.0710	0.4034
52	53.88	39.18	.01708	1.856	33.68	202.49	.0725	.4025
54	55.83	41.13	.01712	1.794	34.44	202.71	.0740	.4017
56	57.83	43.13	.01716	1.735	35.19	202.91	.0754	.4008
58	59.88	45.19	.01720	1.679	35.95	203.13	.0769	.3999
60	62.00	47.30	0.01724	1.624	36.71	203.33	0.0784	0.3991
62	64.17	49.47	.01728	1.572	37.47	203.54	.0798	.3983
64	66.39	51.70	.01732	1.522	38.23	203.74	.0813	.3974
66	68.67	53.98	.01736	1.473	39.00	203.95	.0827	.3966
68	71.01	56.32	.01740	1.427	39.76	204.15	.0842	.3958
70	73.41	58.71	0.01744	1.382	40.52	204.34	0.0856	0.3950
72	75.86	61.17	.01748	1.339	41.29	204.53	.0870	.3941
74	78.37	63.68	.01752	1.298	42.06	204.72	.0885	.3933
76	80.94	66.25	.01756	1.258	42.82	204.90	.0899	.3925
78	83.57	68.87	.01760	1.220	43.59	205.09	.0913	.3918
80	86.26	71.56	0.01764	1.183	44.36	205.27	0.0928	0.3910
82	89.01	74.31	.01768	1.148	45.13	205.45	.0942	.3902
84	91.82	77.13	.01773	1.114	45.90	205.62	.0956	.3894
86	94.70	80.00	.01778	1.081	46.67	205.80	.0970	.3887
88	97.64	82.94	.01782	1.049	47.44	205.96	.0984	.3879
90	100.6	85.95	0.01786	1.018	48.21	206.13	0.0998	0.3872
92	103.7	89.02	.01791	.9889	48.99	206.30	.1012	.3867
94	106.9	92.16	.01796	.9603	49.77	206.46	.1026	.3857
96	110.1	95.37	.01800	.9333	50.54	206.62	.1041	.3850
98	113.4	98.65	.01804	.9069	51.32	206.78	.1055	.3843
100	116.7	102.0	0.01808	0.8814	52.09	206.94	0.1069	0.3836
120	154.2	139.5	.01859	.6710	59.93	208.39	.1206	.3768
140	199.6	184.9	.01915	.5189	67.87	209.58	.1341	.3705
160	253.5	238.8	.01978	.4070	75.90	210.56	.1473	.3646

* Inches of mercury below one atmosphere.

Table 28. Methyl Chloride—Properties of Superheated Vapor

Temp F	Pressure 8 psia (Sat temp −34.5 F)			Pressure 12 psia (Sat temp −19.0 F)			Pressure 16 psia (Sat temp −7.2 F)			Pressure 20 psia (Sat temp 2.5 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(11.06)*	*(191.44)*	*(0.4504)*	*(7.687)*	*(193.64)*	*(0.4399)*	*(6.804)*	*(195.28)*	*(0.4326)*	*(4.710)*	*(197.68)*	*(0.4270)*
0	12.01	197.58	0.4643	7.943	197.06	0.4476	5.907	196.54	0.4355			
10	12.29	199.42	.4682	8.131	198.94	.4516	6.050	198.45	.4396	4.801	197.95	.4300
20	12.57	201.26	.4721	8.318	200.82	.4555	6.192	200.37	.4435	4.917	199.90	.4341
30	12.84	203.12	.4760	8.503	202.70	.4594	6.333	202.27	.4474	5.032	201.82	.4380
40	13.11	204.97	.4798	8.688	204.58	.4632	6.473	204.18	.4513	5.146	203.75	.4420
50	13.39	206.90	0.4836	8.873	206.51	0.4670	6.613	206.10	.4552	5.260	205.71	0.4458
60	13.66	208.82	.4873	9.056	208.43	.4708	6.753	208.85	0.4589	5.373	207.66	.4496
65	13.79	209.79	.4891	9.147	209.41	.4726	6.823	209.04	.4608	5.429	208.66	.4515
70	13.93	210.76	.4910	9.239	210.39	.4745	6.893	210.03	.4627	5.486	209.66	.4534
80	14.21	212.70	.4946	9.422	212.35	.4782	7.033	212.00	.4664	5.599	211.65	.4572
90	14.48	214.68	.4982	9.605	214.34	.4818	7.173	214.01	.4701	5.711	213.67	.4608
100	14.75	216.66	0.5018	9.788	216.34	0.4855	7.311	216.01	.4737	5.823	215.69	0.4645
110	15.02	218.68	.5054	9.971	218.37	.4890	7.449	218.06	0.4773	5.935	217.75	.4681
120	15.29	220.70	.5089	10.15	220.40	.4926	7.587	220.10	.4809	6.046	219.80	.4717
130	15.56	222.75	.5124	10.34	222.47	.4961	7.725	222.18	.4844	6.157	221.90	.4753
140	15.83	224.81	.5159	10.52	224.54	.4996	7.863	224.26	.4879	6.268	223.99	.4788
150	16.10	226.90	0.5193	10.70	226.64	0.5030	8.001	226.83	.4914	6.379	226.12	0.4823
160	16.37	228.99	.5227	10.88	228.74	.5065	8.137	228.49	0.4949	6.489	228.34	.4858
170	16.64	231.12	.5261	11.06	230.88	.5099	8.273	230.64	.4983	6.599	230.40	.4892
180	16.91	233.25	.5295	11.24	233.02	.5133	8.409	232.79	.5017	6.709	232.56	.4927
190	17.18	235.41	.5329	11.42	235.19	.5166	8.545	234.97	.5051	6.819	234.75	.4961

Temp F	Pressure 30 psia (Sat temp 21.4 F)			Pressure 40 psia (Sat temp 35.9 F)			Pressure 50 psia (Sat temp 47.8 F)			Pressure 60 psia (Sat temp 58.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(3.224)*	*(199.00)*	*(0.4169)*	*(2.461)*	*(200.73)*	*(0.4097)*	*(1.992)*	*(202.09)*	*(0.4043)*	*(1.676)*	*(203.17)*	*(0.3998)*
60	3.532	206.70	.4324	2.608	205.69	.4196	2.054	204.65	.4049	1.684	203.49	.4008
65	3.571	207.71	.4343	2.638	206.73	.4216	2.079	205.71	.4114	1.705	204.60	.4028
70	3.610	208.73	.4362	2.668	207.77	.4236	2.104	206.77	.4134	1.727	205.71	.4049
80	3.687	210.77	.4400	2.729	209.84	.4275	2.154	208.59	.4174	1.770	207.92	.4089
90	3.764	212.82	.4438	2.788	211.94	.4313	2.203	211.03	.4213	1.812	210.11	.4129
100	3.840	214.87	0.4475	2.847	214.03	0.4351	2.252	213.18	0.4252	1.854	212.30	0.4169
110	3.916	216.96	.4512	2.906	216.17	.4389	2.300	215.35	.4290	1.895	214.50	.4208
120	3.992	219.06	.4548	2.964	218.32	.4426	2.348	217.62	.4328	1.936	216.69	.4246
130	4.067	221.18	.4585	3.022	220.46	.4463	2.396	219.70	.4366	1.977	218.93	.4283
140	4.142	223.30	.4620	3.080	222.60	.4499	2.443	221.88	.4402	2.017	221.16	.4322
150	4.217	225.45	0.4656	3.138	224.78	0.4535	2.490	224.10	0.4439	2.057	223.41	0.4359
160	4.292	227.61	.4691	3.195	226.96	.4570	2.537	226.32	.4475	2.097	225.66	.4396
170	4.367	229.79	.4726	3.252	229.18	.4606	2.584	228.55	.4511	2.137	227.92	.4432
180	4.442	231.97	.4761	3.309	231.38	.4641	2.630	230.79	.4546	2.176	230.19	.4468
190	4.517	234.19	.4795	3.366	233.62	.4675	2.676	233.05	.4581	2.215	232.48	.4505
200	4.591	236.40	0.4829	3.423	235.86	0.4710	2.722	235.32	0.4616	2.254	234.77	0.4538
210	4.665	238.65	.4862	3.479	238.13	.4744	2.768	237.61	.4650	2.293	237.08	.4573
220	4.739	240.90	.4896	3.535	240.40	.4778	2.813	239.90	.4684	2.331	239.40	.4608
230	4.813	243.20	.4929	3.591	242.72	.4811	2.853	242.24	.4718	2.369	241.75	.4642
240	4.887	245.50	.4962	3.647	245.04	.4845	2.903	244.58	.4752	2.407	244.11	.4676

Temp F	Pressure 80 psia (Sat temp 75.3 F)			Pressure 100 psia (Sat temp 89.6 F)			Pressure 120 psia (Sat temp 101.9 F)			Pressure 140 psia (Sat temp 112.8 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(1.272)*	*(205.84)*	*(0.3928)*	*(1.025)*	*(206.11)*	*(0.3872)*	*(0.858)*	*(208.10)*	*(0.3827)*	*(0.838)*	*(207.91)*	*(0.3790)*
150	1.516	222.00	0.4229	1.191	220.55	0.4124	0.974	219.04	0.4034	0.818	217.48	0.3954
160	1.547	224.32	.4267	1.217	222.94	.4163	0.996	221.51	.4074	.838	220.03	.3995
170	1.578	226.64	.4304	1.243	225.33	.4201	1.019	223.97	.4113	.858	222.56	.4036
180	1.608	228.96	.4341	1.268	227.71	.4239	1.041	226.42	.4152	.877	225.09	.4076
190	1.638	231.30	.4377	1.293	230.10	.4276	1.062	228.88	.4190	.896	227.61	.4115
200	1.668	233.65	0.4413	1.318	232.50	0.4312	1.083	231.33	0.4228	.915	230.13	.4154
210	1.698	236.01	.4448	1.343	234.91	.4349	1.104	233.79	.4265	.934	232.65	.4192
220	1.728	238.37	.4483	1.367	237.32	.4384	1.125	236.25	.4301	.953	235.17	.4229
230	1.758	240.77	.4518	1.391	239.76	.4420	1.146	238.74	.4337	.971	237.70	.4266
240	1.788	243.16	.4552	1.415	242.20	.4455	1.167	241.22	.4373	.989	240.23	.4302
250	1.817	245.56	0.4586	1.439	244.63	0.4489	1.187	243.69	0.4408	1.007	242.73	0.4338
260	1.846	247.95	.4620	1.463	247.06	.4523	1.207	246.16	.4443	1.024	245.24	.4373
270	1.875	250.37	.4654	1.487	249.51	.4557	1.227	248.64	.4477	1.042	247.76	.4407
280	1.904	252.79	.4687	1.511	251.96	.4591	1.247	251.12	.4511	1.060	250.28	.4442
290	1.933	255.24	.4720	1.534	254.44	.4624	1.267	253.64	.4545	1.077	252.82	.4476
300	1.961	257.69	0.4752	1.557	256.92	0.4657	1.287	256.15	0.4578	1.094	255.36	0.4510
310	1.989	260.17	.4784	1.580	259.43	.4689	1.307	258.67	.4611	1.111	257.91	.4543
320	2.017	262.64	.4816	1.603	261.93	.4722	1.327	261.20	.4643	1.128	260.46	.4576
330	2.045	265.14	.4848	1.626	264.45	.4754	1.346	263.75	.4676	1.145	263.04	.4609
340	2.073	267.64	.4879	1.649	266.97	.4786	1.365	266.29	.4708	1.162	265.61	.4641

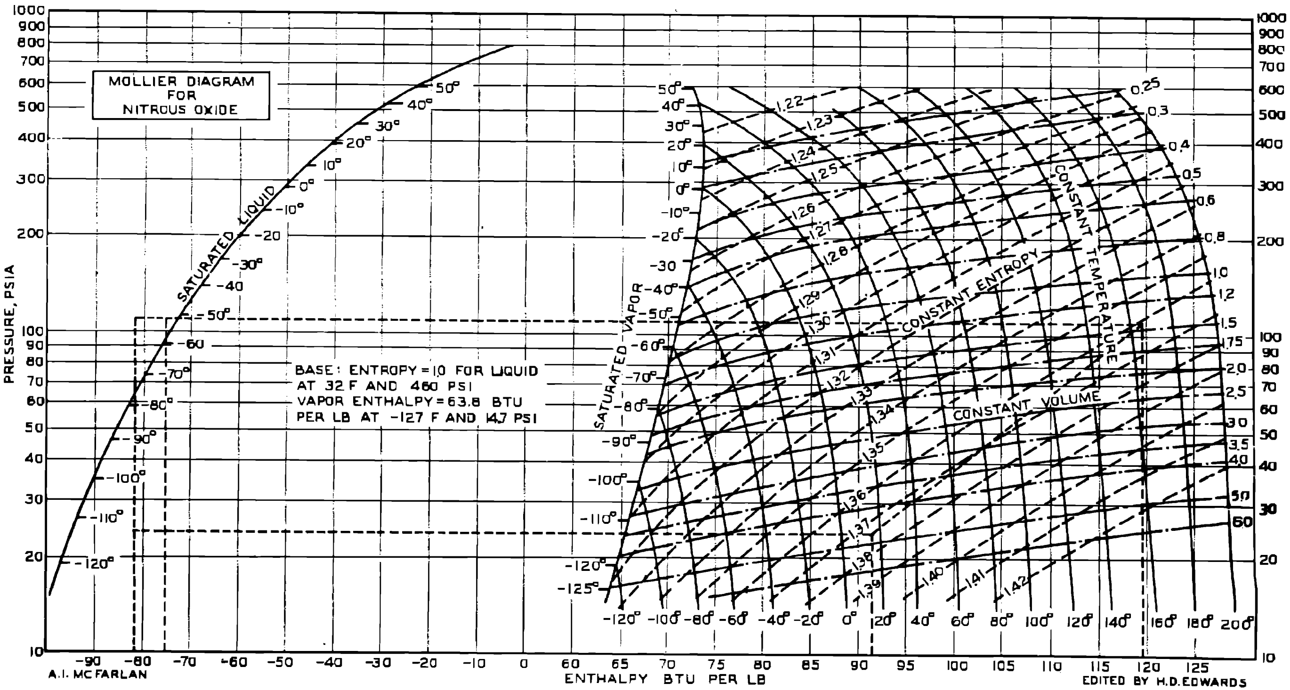
MOLLIER DIAGRAM FOR NITROUS OXIDE
PRESSURE, PSIA
1000
900
800
700
600
500
400
300
200
100
90
80
70
60
50
40
30
20
10
ENTHALPY BTU PER LB
-90
-80
-70
-60
-50
-40
-30
-20
-10
0
60
65
70
75
80
85
90
95
100
105
110
115
120
125
SATURATED LIQUID
50°
40°
30°
20°
10°
0°
-10°
-20
-30°
-40
-50°
-60
-70°
-80°
-90°
-100°
-110°
-120°
SATURATED VAPOR
-125°
CONSTANT ENTROPY
1.22
1.23
1.24
1.25
1.26
1.27
1.28
1.29
1.30
1.31
1.32
1.33
1.34
1.35
1.36
1.37
1.38
1.39
1.40
1.41
1.42
CONSTANT TEMPERATURE
-120°
-100°
-80°
-60°
-40°
-20°
0°
20°
40°
60°
80°
100°
120°
140°
160°
180°
200°
CONSTANT VOLUME
0.25
0.3
0.4
0.5
0.6
0.8
1.0
1.2
1.5
1.75
2.0
2.5
3.0
3.5
4.0
5.0
6.0
BASE: ENTROPY = 1.0 FOR LIQUID AT 32 F AND 460 PSI
VAPOR ENTHALPY = 63.8 BTU PER LB AT -127 F AND 14.7 PSI
A.I. MCFARLAN
EDITED BY H.D.EDWARDS

Chart 15

Table 29. Nitrous Oxide—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum 80 F Btu per lb		Entropy, datum 80 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−127	14.7	0.0	80.0	5.5	−98.5	63.8		1.33
−120	19.0	4.3	79.0	4.25	−96.6	64.5		1.312
−110	26.0	11.3	77.5	3.18	−93.8	65.7		1.3105
−100	35.0	20.3	76.0	2.38	−90.7	66.7		1.3029
− 90	46.0	31.3	75.0	1.82	−87.2	68.0		1.2966
− 80	58.0	42.3	74.0	1.43	−83.7	68.9		1.2887
− 70	74.0	59.3	72.5	1.15	−79.8	70.0		1.2835
− 60	91.0	76.3	70.5	0.935	−75.8	70.7		1.2775
− 50	112.0	97.3	69.3	0.752	−71.8	71.5		1.2722
− 40	137.0	122.3	68.0	0.606	−67.5	72.2		1.2657
− 30	167.0	152.3	67.0	0.513	−63.2	72.7		1.2595
− 20	203.0	188.3	65.5	0.425	−58.7	73.2		1.2530
− 10	240.0	225.3	64.7	0.364	−54.2	73.5		1.2455
0	283.0	268.3	63.1	0.303	−49.4	73.6		1.2365
10	335.0	320.3	61.2	0.262	−44.8	73.7		1.2300
20	387.0	372.3	59.2	0.217	−39.8	73.6		1.2243
32	460.0	447.3	57.0	0.1785	−33.3	73.5	1.00	1.2180
40	520.0	505.3	54.7	0.160	−28.2	73.3		
50	590.0	575.3	52.3	0.138	−21.1	72.6		
60	675.0	660.3	49.2	0.119	−14.2	72.0		
70	760.0	745.3	46.5	0.106	− 7.0	70.7		
80	865.0	850.3	40.0	0.08	0.0	69.0		
97	1069.0	1054.3	26.5	0.0377	66.5	66.5		

Table 30. Specific Volume of Aqua-Ammonia Solutions*

Cu ft per lb of subcooled or saturated liquid

	Weight Concentration Ammonia in Water											
Temp F	0.00	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.00	Temp F
20	.0160	.0166	.0172	.0177	.0182	.0188	.0196	.0206	.0218	.0232	.0247	20
40	.0160	.0167	.0172	.0178	.0183	.0190	.0199	.0209	.0222	.0236	.0253	40
60	.0160	.0167	.0173	.0179	.0185	.0192	.0201	.0213	.0226	.0242	.0260	60
80	.0161	.0168	.0174	.0180	.0186	.0195	.0204	.0216	.0231	.0248	.0267	80
100	.0161	.0169	.0175	.0182	.0189	.0197	.0208	.0221	.0236	.0254	.0275	100
120	.0162	.0170	.0176	.0184	.0191	.0200	.0212	.0226	.0242	.0262	.0284	120
140	.0163	.0171	.0178	.0186	.0194	.0204	.0216	.0230	.0249	.0269	.0294	140
160	.0164	.0172	.0180	.0188	.0197	.0207	.0223	.0237	.0256	.0279	.0306	160
180	.0165	.0174	.0182	.0191	.0200	.0212	.0226	.0244	.0265	.0289	.0320	180
200	.0166	.0175	.0184	.0193	.0204	.0216	.0232	.0252	.0275	.0304	.0338	200
220	.0168	.0177	.0186	.0197	.0210	.0225	.0242	.0264	.0288	.0320	.0361	220

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* Prepared by B. H. Jennings and F. P. Shannon.

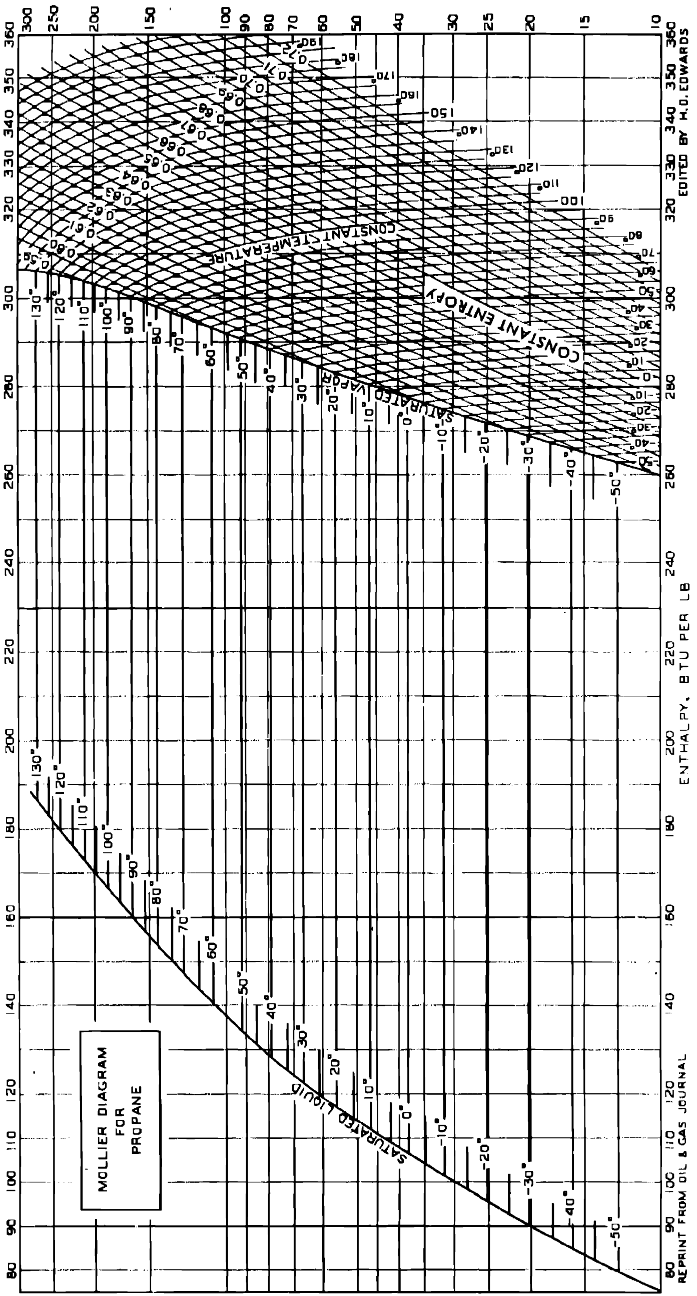
MOLLIER DIAGRAM FOR PROPANE
ENTHALPY, BTU PER LB
80 90 100 110 120 140 160 180 200 220 240 260 280 300 320 330 340 350 360
PRESSURE - P S I A
10 15 20 25 30 40 50 60 70 80 90 100 150 200 250 300
CONSTANT TEMPERATURE
CONSTANT ENTROPY
SATURATED VAPOR
SATURATED LIQUID
130° 120° 110° 100° 90° 80° 70° 60° 50° 40° 30° 20° 10° 0° -10° -20° -30° -40° -50°
EDITED BY H.D. EDWARDS
REPRINT FROM OIL & GAS JOURNAL

Chart 16

Table 31. Propane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy,** datum −200 F Btu per lb		Entropy,** datum −200 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−75	6.37	17.0*	37.59	14.5	65.5	255.5	0.142	0.624
−70	7.37	14.9*	37.40	12.9	68.0	257.0	.148	.623
−65	8.48	12.7*	37.20	11.3	71.5	258.0	.154	.621
−60	9.72	10.1*	37.00	9.93	74.0	259.5	.160	.620
−55	11.1	7.3*	36.80	8.70	77.0	261.0	.167	.618
−50	12.6	4.3*	36.60	7.74	79.5	262.7	0.173	0.617
−45	14.4	0.6*	36.39	6.89	82.3	264.2	.179	.615
−40	16.2	1.5	36.19	6.13	85.0	265.8	.185	.614
−35	18.1	3.4	35.99	5.51	87.5	267.2	.190	.613
−30	20.3	5.6	35.78	4.93	90.2	268.9	.196	.612
−25	22.7	8.0	35.58	4.46	92.8	270.3	0.202	0.610
−20	25.4	10.7	35.37	4.00	95.6	271.8	.208	.608
−15	28.3	13.6	35.16	3.60	98.3	273.2	.214	.607
−10	31.4	16.7	34.96	3.26	101.0	274.9	.220	.607
− 5	34.7	20.0	34.75	2.97	103.8	276.2	.226	.606
0	38.2	23.5	34.54	2.71	106.2	277.7	0.231	0.605
+ 5	41.9	27.2	34.33	2.48	108.8	279.0	0.236	0.604
10	46.0	31.3	34.12	2.27	113.3	280.5	.246	.603
15	50.6	35.9	33.90	2.07	114.0	281.8	.248	.602
20	55.5	40.8	33.67	1.90	116.8	283.1	.254	.601
25	60.9	46.2	33.43	1.74	119.7	284.4	.260	.600
30	66.3	51.6	33.20	1.60	122.3	285.7	0.266	0.599
35	72.0	57.3	32.97	1.48	125.0	287.0	.272	.598
40	78.0	63.3	32.73	1.37	128.0	288.3	.278	.597
45	84.6	69.9	32.49	1.27	131.1	289.5	.285	.596
50	91.8	77.1	32.24	1.18	134.2	290.7	.292	.596
55	99.3	84.6	32.00	1.10	137.2	292.0	0.298	0.596
60	107.1	92.4	31.75	1.01	140.6	293.2	.306	.595
65	115.4	100.7	31.50	0.945	143.8	294.5	.313	.594
70	124.0	109.3	31.24	.883	147.5	295.8	.321	.594
75	133.2	118.5	30.97	.825	150.3	296.9	.327	.594
80	142.8	128.1	30.70	0.770	154.0	298.1	0.335	0.593
85	153.1	138.4	30.42	.722	157.0	299.2	.342	.593
90	164.0	149.0	30.15	.673	160.3	300.3	.349	.593
95	175.0	160.0	29.87	.632	163.4	301.3	.356	.592
100	187.0	172.0	29.58	.591	166.8	302.4	.363	.592
105	200.0	185.0	29.27	0.553	169.8	303.2	0.370	0.592
110	212.0	197.0	28.96	.520	172.8	304.0	.376	.591
115	226.0	211.0	28.63	.488	176.2	304.7	.383	.590
120	240.0	225.0	28.30	.459	179.8	305.2	.391	.589
125	254.0	239.0	27.97	.432	183.5	305.8	.399	.588
130	272.0	257.3	27.64	0.404	186.8	306.1	0.406	0.587
135	288.0	273.3	27.32	.382	190.0	306.3	.413	.586
140	305.0	290.3	27.00	.360	194.0	306.5	.422	.585

Based on material from Dana, Jenkins, Burdick and Timm, published originally in *Refrigerating Engineering*, June 1926, vol. 12, no. 12, page 403.

* Inches of mercury below one standard atmosphere (29.92 in.).

** From Mollier Diagrams for Propane, W. C. Edminster, Standard Oil Co. (Indiana).

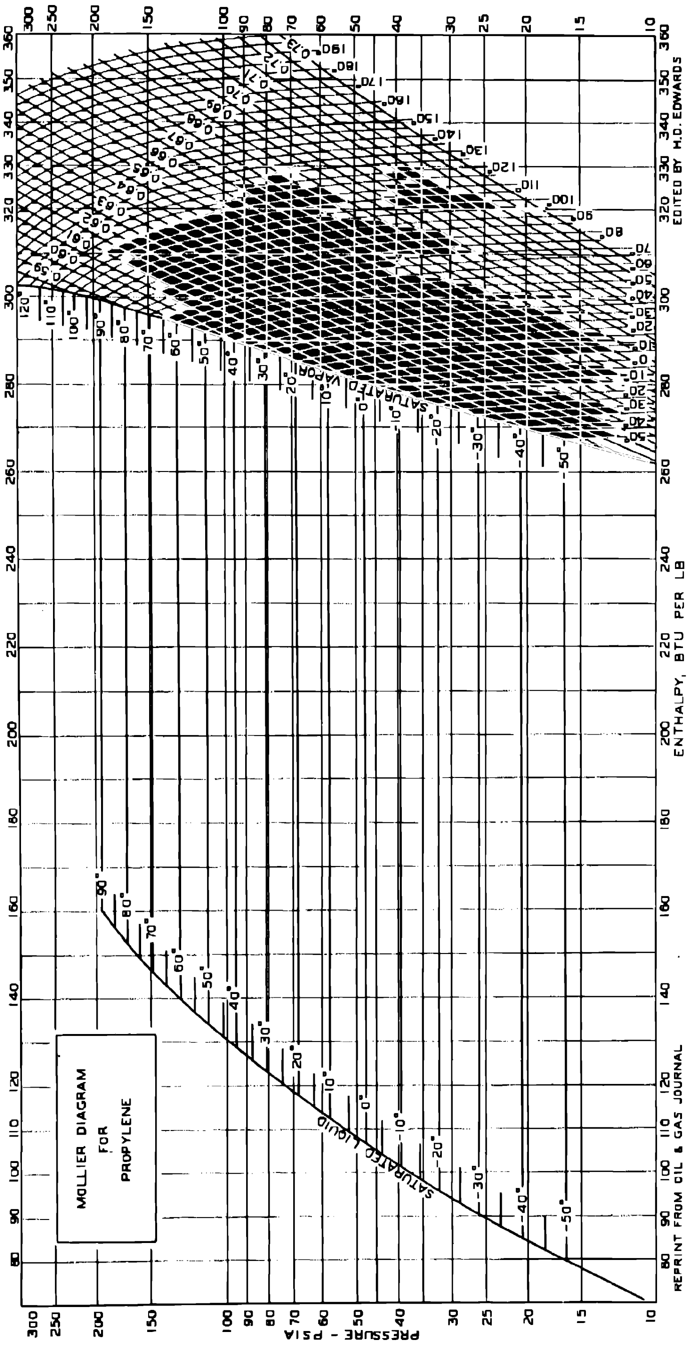
MOLLIER DIAGRAM FOR PROPYLENE
ENTHALPY, BTU PER LB
PRESSURE - PSIA
SATURATED LIQUID
SATURATED VAPOR
REPRINT FROM OIL & GAS JOURNAL
EDITED BY H.D. EDWARDS
80
90
100
110
120
140
160
180
200
220
240
260
280
300
320
330
340
350
360
10
15
20
25
30
40
50
60
70
80
90
100
150
200
250
300

Chart 17

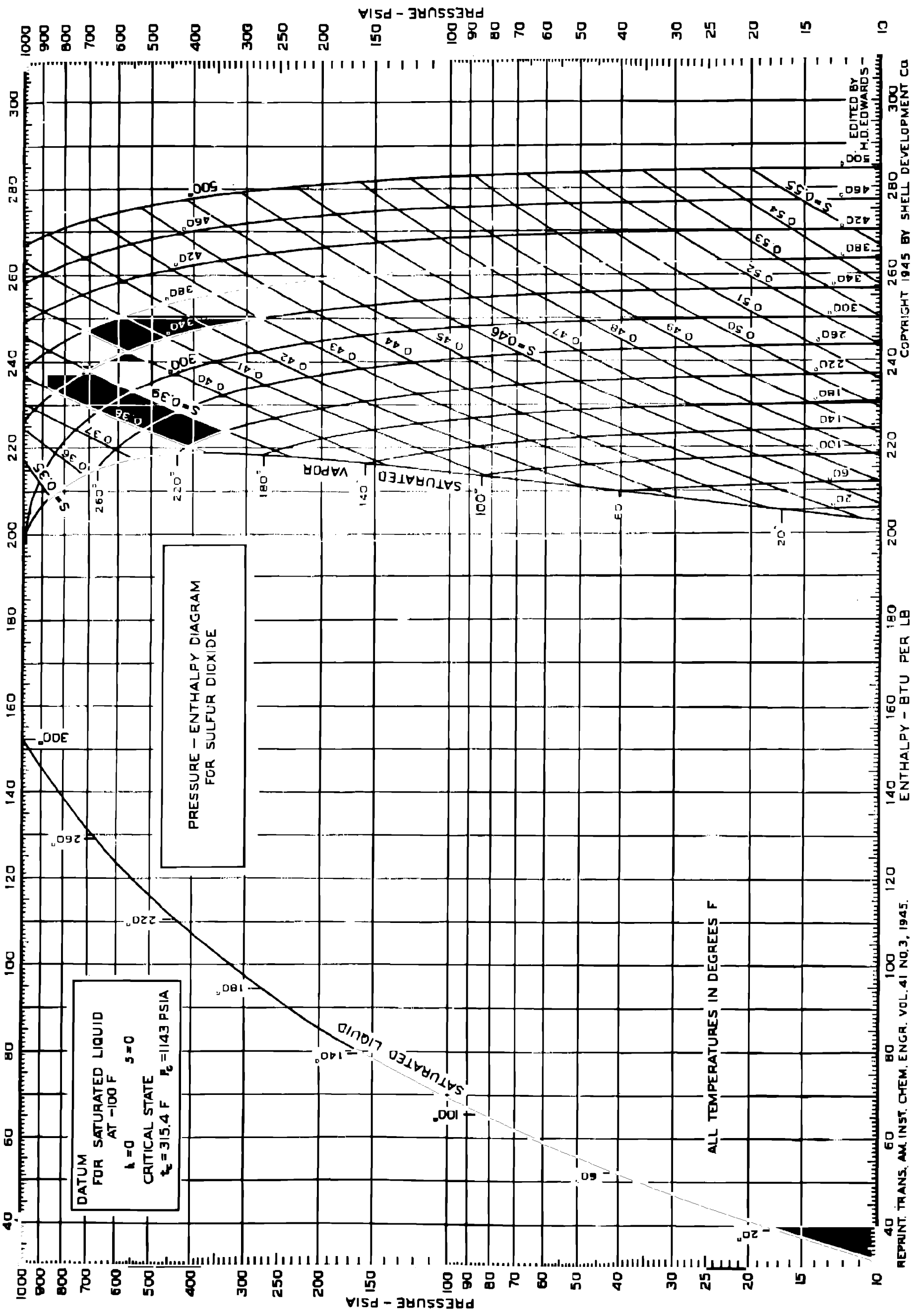
PRESSURE – ENTHALPY DIAGRAM
FOR SULFUR DIOXIDE
DATUM
FOR SATURATED LIQUID
AT -100 F
h = 0
s = 0
CRITICAL STATE
t_c = 315.4 F
P_c = 1143 PSIA
ALL TEMPERATURES IN DEGREES F
SATURATED LIQUID
SATURATED VAPOR
PRESSURE - PSIA
10
15
20
25
30
40
50
60
70
80
90
100
150
200
250
300
400
500
600
700
800
900
1000
ENTHALPY - BTU PER LB
40
60
80
100
120
140
160
180
200
220
240
260
280
300
20°
60°
100°
140°
180°
220°
260°
300°
340°
380°
420°
460°
500°
s = 0.35
s = 0.39
s = 0.46
s = 0.55
0.36
0.37
0.38
0.40
0.41
0.42
0.43
0.44
0.45
0.47
0.48
0.49
0.50
0.51
0.52
0.53
0.54
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Chart 18

Table 32. Sulfur Dioxide—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F		
t	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Evap	Vapor s_g
−40	3.136	23.53*	0.01044	22.42	0.00	178.61	0.00000	0.42562	0.42562
−30	4.331	21.10*	.01053	16.56	2.93	179.90	.00674	.41190	.41864
−20	5.883	17.94*	.01063	12.42	5.98	181.07	.01366	.39826	.41192
−10	7.863	13.91*	.01072	9.44	9.16	182.13	.02075	.38469	.40544
0	10.35	8.85*	0.01082	7.280	12.44	183.07	0.02795	0.37122	0.39917
2	10.91	7.71*	.01084	6.923	13.12	183.25	.02941	.36853	.39794
4	11.50	6.51*	.01086	6.584	13.78	183.41	.03084	.36586	.39670
5†	11.81	5.88*	0.01087	6.421	14.11	183.49	0.03155	0.36454	0.39609
6	12.12	5.25*	.01088	6.266	14.45	183.57	.03228	.36319	.39547
8	12.75	3.97*	.01090	5.967	15.13	183.73	.03373	.36053	.39426
10	13.42	2.61*	0.01092	5.682	15.80	183.87	0.03519	0.35787	0.39306
11	13.77	1.89*	.01093	5.548	16.14	183.94	.03592	.35654	.39246
12	14.12	1.18*	.01094	5.417	16.48	184.01	.03664	.35521	.39185
13	14.48	0.45*	.01095	5.289	16.81	184.07	.03737	.35388	.39125
14	14.84	0.14	.01096	5.164	17.15	184.14	.03808	.35257	.39065
15	15.21	0.51	0.01097	5.042	17.49	184.21	0.03880	0.35125	0.39005
16	15.59	0.89	.01098	4.926	17.84	184.28	.03953	.34993	.38946
17	15.98	1.28	.01099	4.812	18.18	184.34	.04026	.34861	.38887
18	16.37	1.67	.01100	4.701	18.52	184.40	.04098	.34729	.38827
19	16.77	2.07	.01101	4.593	18.86	184.46	.04169	.34598	.38767
20	17.18	2.48	0.01102	4.487	19.20	184.52	0.04241	0.34466	0.38707
21	17.60	2.90	.01104	4.386	19.55	184.58	.04313	.34335	.38648
22	18.03	3.33	.01105	4.287	19.90	184.64	.04385	.34204	.38589
23	18.46	3.76	.01106	4.190	20.24	184.69	.04457	.34073	.38530
24	18.89	4.19	.01107	4.096	20.58	184.74	.04528	.33943	.38471
25	19.34	4.64	0.01108	3.994	20.92	184.79	0.04600	0.33812	0.38412
26	19.80	5.10	.01109	3.915	21.26	184.84	.04671	.33683	.38354
27	20.26	5.56	.01110	3.829	21.61	184.89	.04743	.33553	.38296
28	20.73	6.03	.01112	3.744	21.96	184.94	.04814	.33422	.38236
29	21.21	6.51	.01113	3.662	22.30	184.98	.04886	.33292	.38178
30	21.70	7.00	0.01114	3.581	22.64	185.02	0.04956	0.33163	0.38119
31	22.20	7.50	.01115	3.503	22.98	185.06	.05027	.33034	.38061
32	22.71	8.01	.01116	3.437	23.33	185.10	.05099	.32904	.38003
33	23.23	8.53	.01118	3.355	23.68	185.14	.05171	.32774	.37945
34	23.75	9.05	.01119	3.283	24.03	185.18	.05242	.32645	.37887
35	24.28	9.58	0.01120	3.212	24.38	185.22	0.05312	0.32517	0.37829
40	27.10	12.40	.01126	2.887	26.12	185.37	.05668	.31873	.37541
45	30.15	15.45	.01132	2.601	27.86	185.48	.06020	.31234	.37254
50	33.45	18.75	.01138	2.348	29.61	185.56	.06370	.30599	.36969
55	37.05	22.35	0.01144	2.124	31.36	185.60	0.06715	0.29971	0.36686
60	40.93	26.23	.01150	1.926	33.10	185.59	.07060	.29345	.36405
65	45.13	30.43	.01156	1.749	34.84	185.54	.07401	.28724	.36125
70	49.62	34.92	.01163	1.590	36.58	185.46	.07736	.28110	.35846
75	54.47	39.77	.01169	1.448	38.32	185.34	.08070	.27498	.35568
80	59.68	44.98	0.01176	1.321	40.05	185.17	0.08399	0.26897	0.35296
81	60.77	46.07	.01177	1.297	40.39	185.13	.08462	.26772	.35234
82	61.88	47.18	.01179	1.274	40.73	185.09	.08525	.26652	.35177
83	63.01	48.31	.01180	1.253	41.08	185.05	.08589	.26532	.35121
84	64.14	49.44	.01181	1.229	41.43	185.01	.08653	.26412	.35065
85	65.28	50.58	0.01183	1.207	41.78	184.97	0.08718	0.26291	0.35009
86†	66.45	51.75	.01184	1.185	42.12	184.92	.08783	.26171	.34954
87	67.64	52.94	.01185	1.164	42.46	184.87	.08847	.26052	.34899
88	68.84	54.14	.01187	1.144	42.80	184.82	.08910	.25993	.34843
89	70.04	55.34	.01188	1.124	43.15	184.77	.08974	.25813	.34787
90	71.25	56.55	0.01190	1.104	43.50	184.72	0.09038	0.25693	0.34731
95	77.60	62.90	.01196	1.011	45.20	184.43	.09349	.25103	.34452
100	84.52	69.82	.01204	0.9262	46.90	184.10	.09657	.24516	.34173
110	99.76	85.06	.01219	.7804	50.26	183.31	.10254	.23357	.33611
120	120.93	106.23	.01236	.6598	53.58	182.36	.10829	.22217	.33046
140	158.61	143.91	.01272	.4758	60.04	179.94	.11893	.19990	.31883

* Inches of mercury below one standard atmosphere.

† Standard cycle temperatures.

Note—For values used in making chart No. 18 see Transactions of American Institute of Chemical Engineers, vol. 41, no. 3, June 25, 1945.

Table 33. Sulfur Dioxide—Properties of Superheated Vapor

Temp F	Pressure 10 psia Gage pressure 9.6 in. vac (Sat temp −1.34 F)			Pressure 15 psia Gage pressure 0.3 psig (Sat temp 14.43 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 26.44 F)			Pressure 25 psia Gage pressure 10.3 psig (Sat temp 36.33 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(7.52)*	*(182.95)*	*(0.4000)*	*(5.110)*	*(185.17)*	*(0.39091)*	*(3.878)*	*(184.86)*	*(0.38329)*	*(3.123)*	*(185.26)*	*(0.37754)*
20	7.939	186.7	0.40802	5.192	185.4	0.39270						
30	8.030	188.4	.41159	5.333	187.3	.39672						
40	8.316	190.1	.41505	5.470	189.2	.40054	4.035	187.8	0.38959	3.181	186.1	0.37927
50	8.500	191.8	.41837	5.604	191.0	.40424	4.145	189.8	.39346	3.273	188.4	.38372
60	8.681	193.5	.42161	5.734	192.8	.40777	4.251	191.8	.39719	3.363	190.6	.38795
70	8.860	195.2	0.42480	5.862	195.6	0.41116	4.354	193.7	0.40080	3.451	192.7	0.39198
80	9.038	196.9	.42795	5.988	196.4	.41443	4.454	195.6	.40429	3.536	194.7	.39582
90	9.214	198.6	.43104	6.112	198.2	.41765	4.552	197.5	.40758	3.618	196.7	.39945
100	9.389	200.3	.43407	6.233	199.9	.42076	4.648	199.3	.41093	3.696	198.6	.40291
110	9.563	202.0	.43705	6.353	210.6	.42383	4.742	201.1	.41415	3.772	200.5	.40625
120	9.736	203.7	0.43997	6.471	203.3	0.42682	4.834	202.9	0.41726	3.848	202.4	0.40949
130	9.908	205.4	.44283	6.588	205.6	.42976	4.925	204.7	.42027	3.923	204.2	.41261
140	10.08	207.1	.44565	6.705	206.7	.43264	5.015	206.5	.42322	3.998	206.0	.41568
150	10.25	208.8	.44842	6.821	208.4	.43548	5.104	208.2	.42613	4.073	207.8	.41866
160	10.42	210.5	.45116	6.937	210.1	.43825	5.193	209.9	.42898	4.145	209.6	.42158
170	10.59	212.2	0.45296	7.052	211.8	0.44097	5.281	211.6	0.43176	4.216	211.4	0.42439
180	10.76	213.8	.45651	7.167	213.5	.44366	5.369	213.3	.43449	4.287	213.2	.42717
190	10.93	215.4	.45913	7.282	215.2	.44630	5.456	215.0	.43716	4.358	215.0	.42988
200	11.10	217.0	.46171	7.396	216.9	.44889	5.542	216.7	.43977	4.428	216.7	.43253
210							5.629	218.4	.44234	4.498	218.4	.43413

Temp F	Pressure 40 psia Gage pressure 25.3 psig (Sat temp 58.83 F)			Pressure 50 psia Gage pressure 35.3 psig (Sat temp 70.40 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 80.29 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 88.97 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*1.970*	*(186.60)*	*(0.36470)*	*(1.577)*	*(185.45)*	*(0.35826)*	*(1.3144)*	*(185.16)*	*(0.35272)*	*(1.125)*	*(184.77)*	*(0.34789)*
100	2.246	196.1	0.38415	1.650	193.9	0.37369	1.448	191.4	0.36403	1.181	187.6	0.35443
110	2.304	198.3	0.38810	1.825	196.4	0.37815	1.500	194.3	0.36906	1.228	191.6	0.36020
120	2.360	200.4	.39183	1.872	198.8	.38234	1.548	197.0	.37375	1.272	194.8	.36545
130	2.413	202.5	.39541	1.917	201.1	.38627	1.590	199.5	.37810	1.313	197.6	.37028
140	2.465	204.6	.39881	1.961	203.3	.38998	1.629	201.9	.38217	1.352	200.3	.37478
150	2.515	206.5	.40209	2.003	205.4	.39353	1.669	204.2	.38603	1.389	202.9	.37897
160	2.565	208.5	0.40525	2.044	207.5	0.39691	1.708	206.5	0.38963	1.424	205.3	0.38291
170	2.614	210.4	.40831	2.084	209.6	.40015	1.744	208.6	.39310	1.457	207.6	.38662
180	2.662	212.3	.41127	2.123	211.6	.40327	1.778	210.7	.39639	1.489	209.9	.39014
190	2.709	214.2	.41416	2.161	213.4	.40628	1.808	212.8	.39956	1.521	212.0	.39348
200	2.755	216.0	.41694	2.199	215.4	.40919	1.842	214.8	.40260	1.551	214.1	.39670
210	2.800	217.9	0.41966	2.237	217.3	0.41200	1.874	216.8	0.40554	1.580	216.1	0.39978
220	2.845	219.7	.42233	2.274	219.2	.41477	1.907	218.7	.40839	1.608	218.1	.40275
230	2.889	221.5	.42494	2.311	221.1	.41748	1.939	220.7	.41118	1.636	220.1	.40564
240	2.933	223.3	.42751	2.347	223.0	.42015	1.970	222.6	.41391	1.664	222.1	.40845
250	2.977	225.1	.43007	2.383	224.9	.42275	2.000	224.5	.41657	1.691	224.1	.41120
260	3.021	227.0	0.43262	2.418	226.7	0.42535	2.032	226.4	0.41917	1.718	226.0	0.41389

Temp F	Pressure 80 psia Gage pressure 65.3 psig (Sat temp 96.88 F)			Pressure 100 psia Gage pressure 85.3 psig (Sat temp 110.15 F)			Pressure 120 psia Gage pressure 105.3 psig (Sat temp 121.52 F)			Pressure 140 psia Gage pressure 125.3 psig (Sat temp 131.64 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(at sat)	*(0.9809)*	*(184.33)*	*(0.34367)*	*(0.7786)*	*(183.30)*	*(0.33603)*	*(0.6430)*	*(182.19)*	*(0.33964)*	*(0.5461)*	*(181.04)*	*(0.32588)*
140	1.163	198.6	0.36819	0.8928	194.6	0.35528	0.7085	190.1	0.34264	0.5734	185.1	0.33089
150	1.199	201.3	0.37270	0.9255	197.9	0.36061	0.7403	193.9	0.34904	0.6055	189.7	0.33777
160	1.232	203.9	.37692	0.9561	200.9	.36558	0.7700	197.4	.35484	0.6345	193.6	.34442
170	1.263	206.4	.38093	0.9848	203.7	.37009	0.7972	200.6	.36012	0.6613	196.3	.35041
180	1.292	208.7	.38461	1.012	206.4	.37431	0.8288	203.7	.36494	0.6861	200.8	.35588
190	1.320	211.0	.38813	1.038	209.0	.37829	0.8470	206.7	.36936	0.7092	204.0	.36088
200	1.347	213.3	0.39150	1.062	211.5	0.38203	0.8699	209.4	0.37348	0.7309	207.1	0.36548
210	1.374	215.5	.39471	1.086	213.8	.38556	0.8916	212.0	.37737	0.7513	210.0	.36976
220	1.400	217.5	.39780	1.109	216.1	.38892	0.9124	214.5	.38104	0.7707	212.7	.37379
230	1.426	219.6	.40079	1.131	218.4	.39214	0.9342	217.0	.38451	0.7892	215.4	.37758
240	1.451	221.6	.40369	1.152	220.5	.39524	0.9515	219.3	.38785	0.8070	217.9	.38118

Table 34. Properties of Liquid and Saturated Vapor of Ethylamine, Methylamine, Methylene Chloride, Methyl Formate

Temp F t	Pressure		Liquid, sp vol	Vapor, sp vol	Enthalpy, datum 0 F Btu per lb		Entropy, datum 0 F Btu per lb F	
	psia	psig	cu ft/lb v_f	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
Ethylamine ($C_2H_5NH_2$)								
−58	.355	29.20*		270.8	−7.82	284.78	−0.0263	0.7094
−40	0.740	28.42*		134.5	0.00	290.93	.0000	.6931
−22	1.408	27.06*		72.87	10.93	296.84	.0253	.6788
− 4	2.546	24.75*		41.71	23.01	303.05	.0498	.6664
5	3.342	23.12*	0.02212	32.32	27.61	306.04	.0619	.6609
23	5.590	18.76*		20.00	38.91	311.92	0.0857	0.6512
41	8.960	11.72*		12.88	50.36	317.70	.1091	.6430
68	16.896	2.20		7.156	68.18	326.50	.1437	.6332
86	24.72	10.02		5.039	80.50	332.46	.1664	.6281
113	41.49	26.79		3.136	99.52	341.44	.1999	.6223
Methylamine (CH_3NH_2)								
−58	1.322	27.24*		98.93	−12.4	374.8	−0.0503	0.9135
−40	2.532	25.14*		56.72	0.0	381.7	.0000	.9092
−22	4.551	20.67*		32.56	12.7	388.4	.0294	.8877
− 4	7.78	14.12*		19.84	25.7	395.0	.0586	.8689
5	10.03	9.89*	0.02273	15.54	32.6	398.5	.0731	.8603
23	15.99	1.29		10.01	45.6	404.3	0.1015	0.8443
41	24.49	9.79		6.796	59.2	410.1	.1292	.8300
68	43.52	28.82		3.985	80.2	418.8	.1698	.8115
86	61.53	46.83		2.962	94.5	424.4	.1963	.8008
113	98.76	84.06		1.867	116.9	432.6	.2360	.7873
Methyl Formate ($C_2H_4O_2$)								
0	1.50	26.87*		54.0	0	232.5	0.0000	0.5075
20	2.70	24.43*		31.0	10.3	236.6	.0219	.4934
40	4.66	20.45*		18.9	20.6	240.7	.0432	.4837
60	7.61	14.44*		12.0	30.9	244.8	.0633	.4748
80	12.07	24.55*		7.98	41.2	248.8	.0825	.4670
100	18.26	3.56		5.38	51.5	252.9	0.1015	0.4615
120	27.24	12.54		3.74	61.8	257.0	.1192	.4559
140	38.41	23.71		2.65	72.1	261.0	.1375	.4525
Methylene Chloride (Carrene) (CH_2Cl_2)								
10	1.38	27.12*		42.55	3.4	164.4	0.0072	0.3502
20	1.92	26.01*		31.40	6.8	165.6	.0151	.3461
30	2.56	24.71*		23.90	10.2	166.9	.0222	.3425
40	3.38	23.04*		18.60	13.6	168.0	.0285	.3377
60	5.52	18.72*		11.68	20.4	170.1	.0410	.3292
80	8.81	12.02*		7.50	27.2	172.0	0.0520	0.3202
100	13.25	2.97*		5.14	34.0	173.7	.0620	.3113
120	19.20	4.50		3.65	40.8	175.0	.0714	.3031
140	26.79	12.09		2.69	47.6	176.0	.0795	.2935

* Inches of mercury below one standard atmosphere.
See Table 6.

8. REFRIGERANT CHARACTERISTICS

IN THE early days of mechanical refrigeration, few chemical compounds were available that were suitable as refrigerants and the types of refrigerating equipment were necessarily limited to those that could be used with those few refrigerants. As the industry grew, as new uses for refrigeration appeared, and as different types of equipment were designed, the desirability of new refrigerants more suitable to those uses and equipment designs became more evident.

The use of mechanical refrigeration in the home and in retail businesses called for a low-pressure refrigerant to permit lighter, smaller, and lower priced equipment; sulfur dioxide and later ethyl chloride and methyl chloride made their appearance. The need for a refrigerant having high vapor density, low ratio of compression, and other characteristics suitable for a centrifugal compressor led to the introduction of methylene chloride.

Finally, physical chemists exploring the field of the hydrocarbons of the methane and ethane series, found that by juggling the atoms of these hydrocarbons, substituting chlorine and/or fluorine atoms for hydrogen atoms, they could synthesize almost any type of refrigerant needed for a specific use or for a specific type of equipment. These halogenated hydrocarbons are supplied to the industry under the trade name Freon.

And yet the search goes on, with the trend toward non-flammable, non-toxic, odorless refrigerants of greater stability, even though these characteristics may be obtained at the expense of efficiency, horsepower-per-ton, and coefficient of performance, for these differ so little among the commonly used refrigerants that considerations of safety, stability, boiling point, condensing pressure and other properties suitable to a specific application are apt to be more compelling, especially for the smaller, automatic equipment. In the larger installations attended by operating personnel, the factors affecting operating costs and the size and original cost of the equipment, including piping, may outweigh toxicity, flammability, odor, and even corrosion factors. No one refrigerant may be considered as a universal ideal, and many angles must be considered in determining the suitability of a refrigerant for a specific application or for a specific type of equipment.

Refrigerant **suitability** for a specific application or use should be determined by consideration of (1) thermodynamic, physical and chemical properties; (2) reaction with moisture; (3) leak detection; (4) action on materials; (5) safety considerations and codes; (6) cost; (7) type of compressor; and (8) methods of handling.

I. Thermodynamic, Physical and Chemical Properties (Tables 1, 2 and 3)

1. Boiling temperature and pressure. Inasmuch as the evaporator temperature determines the evaporator pressure, it is desirable to select a refrigerant whose saturation pressure at minimum evaporator operating temperature is slightly above zero gage, in order to maintain a small positive differential between suction (and crankcase) pressure, and atmospheric pressure. Probability of leakage of refrigerant from the system or of air and/or moisture into the low-pressure side of the system, especially at the compressor shaft seal, is less with the smaller pressure differentials. This factor will require less consideration if the motor-compressor unit is of the hermetically-sealed type, but cannot be ignored even in the sealed construction.

2. Freezing temperature. The refrigerant used must have a freezing temperature well below the minimum temperature at

Table 1. Comparative Refrigerant Characteristics

Performance based on 5 F evaporation and 86 F condensation

Refrigerant	Chemical symbol	Molecular weight	Boiling temperature at 0 psig	Freezing temperature at 0 psig	Critical temperature	Critical pressure	Evaporator pressure at 5 F	Condensing pressure at 86 F	Ratio of compression 86 F, 5 F psia	Net refrigerating effect of liquid 86 F, 5 F	Refrigerant circulated per ton 86 F, 5 F	Liquid circulated per ton 86 F, 5 F	Specific volume of vapor, 5 F	Compressor displacement per ton 86 F, 5 F	Horsepower per ton 86 F, 5 F	Coefficient of performance 86 F, 5 F	Temperature of compressor discharge F
			F	F	F	psia	psig	psig	psia	Btu/lb	Lb/min	Cu in./min	cu ft/lb	cfm			
Air	—	—	−318.0	—	−221.0	547.0	—	—	5.00	28.5	7.02	—	11.7	82.3	2.82	1.67	—
Methane	CH_4	16.03	−258.9	−297.0	−115.8	673.0	—	—	5.00	53.8	3.71	—	21.2	78.7	1.203	3.91	—
Ethylene	C_2H_4	28.03	−155.0	−272.0	48.85	731.8	404.3	—	5.00	45.0	4.45	—	.275	1.4	1.532	2.57	—
Nitrous oxide	N_2O	44.02	−127.0	−152.0	96.5	1050.0	318.3	915.3	2.80	85.2	2.35	71.2	.308	.73	1.31	3.59	—
Ethane	C_2H_6	30.04	−127.5	−278.0	90.1	708.3	221.3	661.1	2.87	58.6	3.41	409.0	.533	1.82	2.18	2.16	—
Carbon dioxide	CO_2	44.00	−109.3	− 69.9	87.8	1071.0	316.8	1028.3	3.15	55.5	3.61	167.0	.266	.96	1.827	2.59	160
Propylene	C_3H_6	42.05	− 53.7	−301.0	196.5	667.2	37.0	167.0	4.95	189.2	1.06	—	—	—	1.060	4.46	107
Propane	C_3H_8	44.06	− 44.2	−309.8	202.0	661.5	27.2	140.5	3.70	121.0	1.65	94.0	2.48	4.09	1.020	4.62	97
Freon-22	$CHClF_2$	86.48	− 41.4	−256.0	204.8	716.0	28.3	159.8	4.06	69.3	2.89	68.1	1.25	3.60	1.016	4.65	131
Ammonia	NH_3	17.03	− 28.0	−107.9	271.2	1651.0	19.6	154.5	4.94	474.6	.421	19.6	8.15	3.46	.981	4.75	210
Carrene-7	†	99.29	− 28.0	−254.0	221.1	631.0	16.4	113.4	4.12	61.1	3.27	79.3	1.52	4.97	1.022	4.61	105
Freon-12	CCl_2F_2	120.90	− 21.6	−247.0	233.0	580.0	11.8	93.2	4.08	51.1	3.91	83.8	1.48	5.81	1.005	4.69	102
Methyl chloride	CH_3Cl	50.48	− 10.8	−144.0	289.4	969.0	6.5	80.0	4.48	150.2	1.33	40.9	4.47	5.95	1.020	4.62	178
Isobutane	C_4H_{10}	58.12	10.3	−229.0	272.7	537.0	3.3*	44.8	4.54	111.5	1.79	91.1	6.41	11.50	1.083	4.36	80
Sulfur dioxide	SO_2	64.06	14.0	−103.9	314.8	1141.5	5.9*	51.8	5.63	142.8	1.40	28.6	6.42	9.10	.969	4.86	195
Methylamine	CH_5N	31.05	20.3	−134.5	314.0	1082.0	9.0*	63.2	6.12	304.5	.66	28.2	15.54	1.03	.978	4.81	—
Butane	C_4H_{10}	58.12	31.3	−211.0	306.0	550.1	13.2*	26.9	5.01	123.5	1.62	79.1	9.98	16.16	1.011	4.67	86
Freon-114	$C_2Cl_2F_4$	170.93	38.4	−137.0	294.3	474.0	16.1*	22.0	5.42	43.1	4.64	89.3	4.22	19.59	.997	4.74	71
Freon-21	$CHCl_2F$	102.93	48.0	−211.0	353.3	750.0	19.3*	16.5	5.97	89.4	2.24	45.8	9.12	20.46	.932	5.11	140
Ethyl chloride	C_2H_5Cl	64.52	54.5	−217.7	369.0	764.0	20.5*	12.4	5.83	142.3	1.41	44.4	16.10	22.62	.906	5.19	106
Ethylamine	C_2H_7N	45.06	61.8	−115.0	362.0	815.0	23.1*	10.3	7.25	243.0	.823	323.0	32.32	26.4	.855	5.55	—
Freon-11	CCl_3F	137.38	74.7	−168.0	388.4	635.0	24.0*	3.6	6.24	67.5	2.96	56.0	12.27	36.3	.943	5.00	113
Methyl formate	$C_2H_4O_2$	60.04	89.2	−147.5	418.0	607.0	26.3*	1.8*	7.75	189.2	1.06	29.9	47.25	49.9	—	—	149
Ethyl ether	$C_4H_{10}O$	74.08	94.3	−177.3	522.1	380.8	26.9*	4.9*	8.20	126.3	1.58	62.9	35.00	55.4	.822	5.74	—
Methylene chloride	CH_2Cl_2	84.93	103.6	−142.0	421.0	670.0	27.4*	8.4*	9.07	134.6	1.49	30.9	49.90	74.3	.968	4.87	205
Freon-113	$C_2Cl_3F_3$	187.39	117.6	− 31.0	417.4	495.0	27.9*	13.9*	8.02	53.7	3.73	66.4	27.04	100.9	.981	4.84	81
Dichlorethylene	$C_2H_2Cl_2$	96.93	118.0	− 70.0	470.0	795.0	28.3*	15.8*	8.42	114.3	1.75	38.3	63.6	111.2	.973	4.83	—
Trichlorethylene	C_2HCl_3	131.37	187.0	−124.3	520.0	728.0	29.6*	26.2*	11.65	91.7	2.18	41.6	229.4	502.0	.980	4.82	—
Water (40 F and 86 F)	H_2O	18.02	212.0	32.0	706.1	3226.0	29.7*	28.7*	5.06	1025.3	.195	5.4	2444.0	476.6	1.125	4.20	282

* Inches of mercury vacuum.

† Azeotropic mixture of Genetron-100 (CH_3CHF_2) and Freon-12 (CCl_2F_2).

which the evaporator may possibly operate.

3. **Condensing temperature and pressure.** In installations designed for a specific location with known temperature limits of the condensing medium, the refrigerant to be used may be more accurately chosen, than if, as is generally true, the equipment is made for nation-wide use, and subject to wide temperature limits of the condensing medium. For this latter type of equipment the maximum condensing temperature likely to be encountered must be a governing factor in the selection of the refrigerant.

In any case, it is desirable to use a refrigerant whose condensing pressure at extremes of design and safety conditions is low enough to permit the use of lightweight material in the high-pressure portion of the system, including non-ferrous sheet metals or pipe of light gages. This applies to the condenser, receiver, liquid line, expansion valve, liquid line solenoid, or other accessories subject to condensing pressure. Such construction is desirable from considerations of safety, original and operating costs, and conservation of space.

The condensing pressure will be a determining factor in the choice of type of condenser, air-cooled or water-cooled. Thus it may automatically eliminate a refrigerant from applications in which the air-cooled condenser is a practical and economic necessity.

There is also increased likelihood of leakage of refrigerant from the system and greater danger in handling or working around a refrigerant having a relatively high condensing pressure.

4. **Discharge temperature.** Refrigerants that have relatively high compressor discharge temperatures are apt to cause oil break-down and formation of sludges unless highly refined, expensive oils are used.

5. **Critical temperature.** The refrigerant used must have a critical temperature well above the maximum condensing temperature that may be experienced, even under extreme conditions. The critical temperatures of most of the commonly used refrigerants (except carbon dioxide) are high enough that this factor may be disregarded for most types and locations of equipment.

6. **Critical pressure.** The refrigerant must have a critical pressure well above the likely maximum condensing pressure. High power consumption for compression results if the critical pressure is but little higher than the condensing pressure corresponding to normal water or air temperature. As is true of the critical temperature, this factor is satisfactory for all of the commonly used refrigerants other than carbon dioxide.

7. **Compression ratio.** A refrigerant having a low ratio of compression is desirable as to first cost of the equipment and cost of operation. The original cost of the compressor will be increased by a high ratio of compression because of the higher precision in manufacture necessary to minimize losses from clearance volume and from leakage between the cylinder walls and pistons. As these factors affect volumetric efficiency, the overall cost of operation is correspondingly affected.

High compression ratios affect the design, construction and operating characteristics of expansion valves, liquid line solenoid valves and the like, enabling them to operate at the high difference in pressure on the two sides of the valve.

Rotary and centrifugal type compressors should use only refrigerants having comparatively low ratios of compression to minimize vapor slippage past the rotor.

8. **Latent heat of vaporization.** Other factors being equal, a high latent heat of vaporization is desirable, since it affects the net refrigerating effect, the amount of refrigerant circulated, and the size and cost of piping and equipment. The value of a high latent heat of vaporization is sometimes over-rated, however, for its bearing on the selection of a refrigerant is affected by other important factors, such as specific heat of the liquid, and by the density of the liquid and of the vapor. In fact, the small machine (household and small commercial sizes) favors a refrigerant with a moderate to low latent heat of vaporization, to prevent the expansion valve or refrigerant float valves from having such small orifices as to make very difficult the accurate metering of liquid into the evaporator.

9. **Specific heat of the liquid.** The latent heat of vaporization is the "refrigerating ability" of the refrigerant. It is reduced because some of it is required to cool the

Table 2. Comparative Refrigerant Characteristics at Various Evaporating and Condensing Temperatures

	Evaporating pressure psig	Condensing pressure psig	Ratio of compression	Net refrigerating effect Btu/lb	Refrigerant circulated per ton lb/min	Specific volume of vapor cu ft/lb	Compressor displacement per ton cu ft/min
			−130 F Evaporating, −40 F Condensing				
Ethylene	16.16	196.7	6.8	141.4	1.42	3.72	5.27
Nitrous oxide	1.02″	145.3	11.3			5.94	
Ethane	2.24″	98.2	8.3	155.1	1.29	8.16	10.52
Carbon dioxide	18.84″	131.0	27.1	130.6	1.53	14.74	22.60
Freon-22	28.51″	.61	22.2	89.7	2.25	58.21	130.0
			−76 F Evaporating, 5 F Condensing				
Ethylene	95.0	399.8	3.8	117.0	1.71	1.14	1.95
Nitrous oxide	60.3	318.3	4.4			1.22	
Ethane	40.4	221.3	4.2	131.1	1.53	2.30	3.51
Propane	17.4″	35.9	8.2	147.7	1.35	14.80	20.02
Freon-22	18.87″	28.3	8.1	84.2	2.38	8.56	20.37
Ammonia	23.47″	19.6	10.9	534.7	.37	75.00	28.05
Freon-12	23.54″	11.8	7.8	59.9	3.34	10.02	33.47
Methyl chloride	25.34″	6.5	9.4	169.2	1.18	36.00	42.55
			−40 F Evaporating, 68 F Condensing				
Nitrous oxide	145.3	730.3	4.7			.600	
Ethane	98.2	532.2	4.9	72.8	2.75	1.136	3.12
Carbon dioxide	131.1	816.9	5.7	75.9	2.60	.611	1.59
Propylene (approx)	21.0	145.0	6.9	122.0	1.65		
Propane	1.5	104.9	13.4	168.6	1.19	6.13	7.28
Freon-22	.61	118.3	8.7	70.1	2.85	3.28	9.35
Ammonia	8.7″	109.6	11.9	479.3	.42	24.86	10.38
Carrene-7	7.9″	82.7	9.0	60.4	3.31	4.08	13.50
Freon-12	11.0″	67.5	8.9	50.1	3.99	3.91	15.60
Methyl chloride	15.9″	56.3	10.3	150.9	1.33	12.72	16.85
Sulfur dioxide	23.6″	32.6	14.5	143.4	1.40	22.2	31.00
			20 F Evaporating, 80 F Condensing				
Propane	40.8	128.1	2.57	128.5	1.55	1.900	2.96
Freon-22	43.3	145.0	2.76	72.96	2.74	.937	2.57
Ammonia	33.5	138.3	3.18	485.8	.41	5.910	2.44
Carrene-7	27.3	102.5	2.79	64.9	3.08	1.144	3.52
Freon-12	21.0	84.0	2.76	54.2	3.69	1.121	4.14
Methyl chloride	14.5	71.6	2.96	154.5	1.29	3.312	4.28
Isobutane	3.5	39.2	3.97	119.5	1.67	4.680	7.84
Sulfur dioxide	2.5	45.0	3.48	144.5	1.38	4.487	6.22
Methylamine	.2	40.5	3.71	313.6	.64		
Butane	6.3″	22.9	3.24	133.0	1.51	7.230	10.93
Freon-114	10.1″	18.3	3.44	46.6	4.29	3.000	12.87
			40 F Evaporating, 100 F Condensing				
Propane	63.3	172.0	2.40	121.5	1.65	1.37	2.25
Freon-22	69.0	197.9	2.54	68.1	2.94	.656	1.93
Ammonia	58.6	197.2	2.89	467.8	.428	3.294	1.30
Carrene-7	46.2	141.9	2.57	61.4	3.26	.802	2.61
Freon-12	37.0	116.9	2.54	51.6	3.88	.792	3.07
Methyl chloride	27.9	104.3	2.80	149.4	1.34	2.32	3.11

Table 2. Comparative Refrigerant Characteristics at Various Evaporating and Condensing Temperatures (Continued)

	Evaporating pressure psig	Condensing pressure psig	Ratio of compression	Net refrigerating effect Btu/lb	Refrigerant circulated per ton lb/min	Specific volume of vapor cu ft/lb	Compressor displacement per ton cu ft/min
40 F Evaporating, 100 F Condensing							
Isobutane	12.2	59.0	3.74	111.0	1.80	3.22	5.80
Sulfur dioxide	12.4	69.8	3.12	138.5	1.44	2.89	4.17
Methylamine	9.3	62.8	3.23	304.8	.66		
Butane	3.0	37.5	2.95	128.0	1.56	4.88	7.63
Freon-114	.5	31.7	3.05	44.7	4.47	1.98	8.85
Freon-21	4.8″	25.3	3.28	90.0	2.22	4.13	9.18
Freon-11	15.6″	8.9	3.36	68.8	2.90	5.45	15.8
Methyl formate	20.45″	3.6	3.91	189.2	1.06	18.90	20.0
Methylene chloride	23.04″	2.97″	3.92	134.0	1.49	18.00	27.8
Freon-113	24.52″	8.59″	3.95	55.7	3.60	10.68	38.7
Water	29.67″	27.99″	7.75	1011.3	.198	2438	482.0

″ Inches of mercury vacuum.

warm liquid ahead of the expansion valve down to evaporator temperature. What is left is known as the net refrigerating effect. A low specific heat of the liquid is desirable, for the less the latent heat of vaporization used in cooling the liquid, the more will be left as net refrigerating effect.

10. Density of the liquid. A refrigerant having high liquid density may first appear to be desirable because of savings in size and cost of receiver, liquid line pipe or tubes and other vessels containing liquid refrigerant. Such savings are largely offset, however, by the increased viscosity that parallels increase in density, requiring increased sizes of pipe and orifices to maintain low pressure drops.

In installations in which the evaporator is placed several floors above the condensing unit, the formation of flash gas in the liquid line is a factor that must be considered; the greater the density of the liquid refrigerant the lower the level at which this will occur.

Since refrigerant is sold by the pound, the refrigerant charge for an installation will, other factors being equal, be more for a refrigerant having high liquid density. This factor may be of minor concern in small, self-contained equipment, in which the refrigerant charge is small and the liquid line short and of small diameter.

11. Viscosity of the liquid. Pressure drop in long liquid lines may be sufficient to require larger and more costly piping in order to prevent excess flash gas and reduced capacity of the expansion valve. In large installations these factors may require serious consideration.

12. Specific heat of the vapor. If the heat capacity or specific heat of the vapor is high, the heat that is added to the saturated vapor, superheating it in the evaporator and suction line, will raise its temperature less than if the specific heat is low. As its temperature rises, the superheated vapor expands and its density decreases. This is unfavorable to the compressor capacity, for the cylinders pump more refrigerant vapor if it is saturated, cool and dense than if it is superheated, warmed and less dense. Therefore, a high specific heat of vapor is desirable. The gains effected by use of heat exchangers may offset superheating losses of the compressor only, and result in added efficiency to the system as a whole.

13. Density of the vapor. One might assume that high vapor density would permit the use of smaller tubes, piping and orifices in the pressure side of the system, but the reverse is true, due to high pressure drop resulting from high vapor density. Moderately low vapor densities are, therefore, desirable in ordinary compression systems, except those using centrifugal compressors, for which refrigerants having high vapor densities are desirable.

14. Viscosity of the vapor. To minimize pressure drop and consequently the sizes and costs of tubes and piping, a low vapor viscosity is desirable.

15. Thermal conductivity. An ability to conduct the heat quickly from the evaporator or to the condenser in either the liquid or vapor phases is a desirable characteristic of a refrigerant, since it increases evaporator and condenser performance. Therefore high thermal conductivities of refrigerant vapor or liquid are desirable.

16. Oil solubility. Some refrigerants are highly miscible with compressor oil, consequently picking up oil from the compressor and carrying it through the system. Some oil passing the pistons and discharge valves is desirable for lubrication, and to seal the valve surfaces, yet oil serves no useful purpose in the rest of the system, with the possible exception of some types of refrigerant flow controls.

17. Horsepower per ton. With a theoretically perfect cycle, all refrigerants would be equal as to power required per ton of refrigeration. In actual practice there is very little difference in horsepower per ton between the commonly used refrigerants except in the case of carbon dioxide. Thus this factor is much less important than most of the other characteristics.

In the smaller machines the unit ordinarily used instead of horsepower per ton is Btu per watt, which is of inverse character to horsepower per ton.

18. Stability. Ideally, any refrigerant used in a compression system should be absolutely stable and inert, that is, it should have no chemical reaction with any material—oil, gas (including air), metal, moisture or other substance of any nature with which it might come in contact, nor be chemically affected by any extremes of temperature to which it might be subjected.

19. Dielectric strength of vapor. This characteristic is important only in the "hermetic" units, in which the refrigerant comes into direct contact with the field coils of the motor driving the compressor. If the dielectric strength of the refrigerant vapor at suction pressure and the temperature are low, possibility of short-circuits between electrical conductors is increased, although in most designs no bare conductors are exposed to the refrigerant, all being fully insulated. Table 3 shows the relative dielectric strengths of the vapor at atmospheric pressures of several of the refrigerants that are frequently used in hermetic motor-compressor units.

20. Leakage tendency. The smaller the molecule of the refrigerant, the harder it is to prevent it from leaking. The size of the molecule is proportional to the square root of the molecular weight. Pressure, capillarity, viscosity, density and rate of diffusion are also factors that affect the tendency of a gas to leak, but these being equal, the refrigerant with a high molecular weight has a lower leak tendency.

21. Washing effect. The hydro-carbon refrigerants are all excellent oil and grease solvents, which makes it especially important that castings particularly, but other materials as well, be degreased and very carefully cleansed after machining and fabrication. Otherwise the continual washing of the parts by the refrigerant or refrigerant-oil mixture circulating in the system will dissolve the oils on the inner surfaces of the system or soaked into the pores of castings and will loosen scale, dirt and other impurities which will clog screens, foul orifices and aggravate sludge formation.

The action of the refrigerant on gasket material and the nature of the impregnation of seal surfaces may lead to difficulty in "holding" the refrigerant.

22. Odor. An easily detectable odor has an advantage in leak detection in that maintenance personnel will become aware of leakage and make the necessary repair, or that the user of untended equipment will know when to call the repairman. An odor can serve also as a warning to occupants to avoid the refrigerant. On the other hand, a pronounced odor, especially one of an irritating nature, may cause hysteria and panic in congested public places or institutions.

There are but few refrigerants that have a pronounced odor that is easily detected. These include ammonia and sulfur dioxide, and, to a lesser degree, ethyl chloride and ethane. Most of the hydrocarbons, including the halogens, methyl chloride, methyl-

Table 3. Applications, Leak Methods and Oil Miscibility

	Application for which the refrigerant is used or is suitable							Applicable leak test methods[4,5]	Miscibility with oil	Dielectric strength of vapor[6]
	Industrial with attendance[1]	Commercial up to 25 hp[1]	Air conditioning[1]	Household[1]	Evaporator temperature range[2]	Type of compressor[3]	Air or water-cooled			
Methane	Ltd.	None	None	None	Ult.	Recip.	Water	2, 4	Yes	1.00
Ethylene	Ltd.	Few	None	None	Ult.	Recip.	Water	2, 4	Yes	1.21
Nitrous oxide	Ltd.	Few	Suit.	None	Ult.	Recip.	Water	2, 4	No	1.14
Ethane	Ltd.	Few	None	None	Ult.	Recip.	Water	2, 4	Yes	
Carbon dioxide	Ext.	Few	Ltd.	None	Low & Med.	Recip.	Water	2, 4	No	.88
Propylene	Ltd.	Ltd.	None	None	Ult. & Low	Recip.	Water	2, 4	Yes	
Propane	Ltd.	Ltd.	None	None	Ult. & Low	Recip.	Water	2, 4	Yes	
Freon-22	Ltd.	Ltd.	None	Suit.	Ult. & Low	Recip.	Water	1, 3, 5	Yes	1.31
Ammonia	Ext.	Ext.	Ltd.	Ext.	Low to High	Recip.	Water	1, 3, 6	No	.82
Carrene-7	Suit.	Suit.	Suit.	Suit.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 5	Yes	
Freon-12	Ext.	Ext.	Ext.	Ext.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 5	Yes	2.40
Methyl chloride	Ltd.	Ext.	Ltd.	Ext.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 5	Yes	1.06
Isobutane	Few	Ltd.	None	Ltd.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3	Yes	
Sulfur dioxide	Few	Ext.	Few	Ext.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 6	No	1.90
Methylamine	Few	Suit.	None	Suit.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3	Yes	
Butane	Few	Suit.	None	Ltd.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3	Yes	
Freon-114	None	None	None	Ext.	Med. & High	Rotary	Air	1, 3, 5	Yes	2.80
Freon-21	Few	None	None	None	Med. & High	Rotary	Air	1, 3, 5	Yes	1.33
Ethyl chloride	None	Few	None	Ltd.	Med. & High	Rotary	Air	1, 3, 5	Yes	1.00
Ethylamine	Suit.	None	None	None	High	Rot.-Cen.	Air	1	Yes	
Freon-11	Suit.	Few	Ext.	Ext.	High	Rot.-Cen.	Water	1, 5	Yes	3.00
Methyl formate	Suit.	None	None	None	High	Centrif.	Water	2, 5	Yes	
Ethyl ether	Suit.	None	None	None	High	Centrif.	Water	2	Yes	
Methylene chloride	Suit.	None	Ext.	None	High	Centrif.	Water	2, 5	Yes	1.11
Freon-113	Suit.	None	Ext.	None	High	Centrif.	Water	2	Yes	2.60
Dichlorethylene	None	None	None	None	High	Centrif.	Water	2, 5	Yes	
Trichlorethylene	None	None	None	None	High	Centrif.	Water	2, 5	Yes	
Water	Ltd.	None	Ext.	None	High	Cen.-Jet.	Water	2	No	

[1] Ext.—Extensive; Ltd.—Limited; Few—Few, if any; None—None; Suit.—Suitable but few if any.

[2] Ult.—Ultra-Low, −130 to −75 F; Low—Low, −75 to 0 F; Med.—Medium, 0 F to 30 F; High—High, 30 F to 50 F.

[3] Rec.—Reciprocating; Cen.—Centrifugal; Rot.—Rotary.

[4] Numbers refer to those described on page 195.

[5] When using the Halide torch on equipment charged with USA Group 2 refrigerants (see Table 4) make certain that good ventilation is provided. Do not use Halide torch for USA Group 3 refrigerants.

[6] Ethyl chloride as unity.

ene chloride, dichlorethylene and trichlorethylene, have an odor which is not readily detectable. It is a slightly sweet, ethereal odor similar to that of natural gas. Methyl chloride may be obtained with 1% acrolein in it to give it a pungent, irritating odor.

Carbon dioxide, the Freon group, and water when used as a refrigerant have practically no odor. Also, they are essentially harmless, so no irritating odor is needed to serve as a warning of leakage. The products of decomposition of the nonflammable or slightly flammable halogenated hydrocarbons by a flame, or in some cases by hot wire, have pungent, detectable odors that serve as warning agents against those toxic gases produced by decomposition.

Azeotropes. A comparatively recent development in refrigerants is the azeotropic mixture of two refrigerants. Such a mixture does not resolve into its component refrigerants at change of pressure or temperature, and has its own fixed thermodynamic properties that are unlike either of its components. Several azeotropes are known, but the only one in actual use, is Carrene-7, an azeotropic mixture of 73.8% Freon-12 (CCl_2F_2) and 26.2% Genetron-100 (CH_3CHF_2). It is of particular interest in that its displacement characteristics permit it to be used in direct driven compressors at 50 cycle speed with about the same refrigerating capacity and evaporating temperature, as those same compressors driven at 60 cycle speed, and charged with Freon-12.

Refrigerants and absorbents in an absorption system. A refrigerant for use in an absorption system should, in general, have the same desirable qualities as a refrigerant for a compression system, and in addition it must be suitable for use with an absorbent so selected that in the combination:

(a) The vapor pressures of the two at the generator temperature shall be widely divergent, in order to obtain as nearly complete fractionation as possible in the condenser and rectifier.
(b) Temperature-pressure relations should conform to practical absorber and generator temperatures and pressures.
(c) The refrigerant must have high solubility in the absorbent at the absorber temperature and pressure (as governed by the condensing medium), and low solubility at the generator temperature and pressure.
(d) In most absorption equipment the solvent must remain liquid at absorber and generator temperatures and pressures.
(e) The refrigerant and the solvent must remain stable with one another within the generator-evaporator range of temperatures. The solvent should have low specific heat, surface tension and viscosity, and must be neutral to the materials used in the equipment.

There are many known combinations of refrigerant and absorbent that could be used in an absorption system, but the only ones which have been commonly used are ammonia-water, methylene chloride-dimethyl ether of tetraethylene glycol, water-lithium chloride, water-lithium bromide, ammonia-calcium chloride, ammonia-strontium chloride, and sulfur dioxide-silica gel.

II. Reaction with Moisture

Water in a refrigerating system has two effects, either of which is sufficiently injurious to warrant the utmost care in preventing the entrance of moisture into a system, or in reducing to the lowest practical minimum any moisture that may have been introduced into the system.

1. Water unites in varying degrees with most of the refrigerants, which with, or in addition to, any air that may be in the system, in turn react on the oil and the various materials of which the equipment is made—metals, plastics, fibers, and various compounds in valve packing, gaskets, and electrical insulation. If the refrigerant holds considerable moisture in solution (as do ammonia, sulfur dioxide and carbon dioxide), this moisture is still able to cause **corrosion** to some of the materials with which it comes in contact. Therefore, because no trouble arises from freeze-ups at the expansion valve, it is not necessarily true that any moisture that is in the system is doing no harm. It is not uncommon to open a system and find damaging corrosion, although the system has operated with no expansion valve freeze-up trouble. Corrosion first makes itself known by pitted and damaged valve needles and seats; if not checked, it may extend to seal leakage, and damage to bearings, bushings, pins, shafts and other finely finished surfaces. Rust and scale washed from corroded surfaces may clog screens, filters and valve ports and restrict the flow of refrigerant.

Sludges and precipitates may form as a result of chemical reaction between the refrigerant and the oil, and in the presence of impurities, foreign substances or adulterants, their reactions are aggravated and amplified by moisture and/or air in the system. It is wise to exclude everything from the system other than the refrigerant and the oil of specifications approved by the manufacturer of the equipment, especially the manufacturer of the compressor.

2. After the refrigerant has absorbed all the moisture of which it is capable, any further moisture in the system exists as free water, which freezes into ice in those portions of the system below 32 F. This usually occurs first at the orifice in the expansion valve or float valve or in the capillary tube. **Formation of ice** causes restriction or stoppage of refrigerant flow and refrigeration ceases. The valve then warms up, the ice melts, and refrigeration is resumed. This may reoccur, causing "sticky" operation of the valve, or the water may pass on into the evaporator where it may freeze and remain as small droplets of ice. It may remain frozen in the evaporator for a considerable period until the evaporator is defrosted, when it returns to the water state and again circulates through the condensing unit to the expansion valve. It is, therefore, imperative that moisture be excluded or removed from the compression system.

When refrigerants are warm, they carry higher percentages of moisture than when they are cold. So when a refrigerant that is saturated with moisture is chilled, it releases some of its moisture content which then appears in the system as free water.

For full information on moisture in the system and methods for its removal, see Chapter 18.

III. Leak Detection

The most frequent cause of service calls on refrigerating systems is "leaks in the system," usually caused by:

a. In portions of the system in which the pressure is **above atmospheric pressure,** the refrigerant will leak out of the system, eventually creating a shortage and necessitating replacement of the amount of leakage.

b. In other portions in which the pressure may be **below atmospheric** (on a vacuum), the leakage will be of air into the system. In most climates, air contains enough moisture to cause trouble in time inside the system. In either case the leak must be found and repaired and refrigerant added as required, or air and moisture removed.

The **methods** for **detecting leaks** differ according to the refrigerant used, the size and type of equipment being tested, and the facilities available at the test location.

1. In factories or well equipped shops, a favorite method is to put a pressure of 200 or 300 psi (depending upon the strength of the equipment) of **dried air** or some inert gas such as carbon dioxide or nitrogen, in the equipment and then submerge the equipment in a brightly lighted tank of water or kerosene. Location of a leak will be discovered by a stream of bubbles rising from it. The tank must be brilliantly lighted (preferably from beneath the water surface as well as from above) and the liquid clear; otherwise a fine stream of bubbles from a very small leak may escape detection.

2. Large equipment is frequently tested by putting **nitrogen** or **carbon dioxide** (never oxygen) in it to a definite pressure and letting it stand for at least 24 hr. Constancy of gage readings indicates absence of leaks. Allowance must be made for variations in pressure due to variations in ambient temperature.

3. Another method of testing is by **pumping a vacuum** on the equipment by means of an auxiliary vacuum pump (preferable to using the refrigerating compressor for this purpose), with the pump discharging through a tube, the outlet end of which is submerged in clear oil in a glass vessel. After discontinuance of bubbles, indicating a high degree of vacuum, the equipment is allowed to stand for an hour or two with the vacuum pump idle. The pump is then started. Issuance of bubbles indicates that air has leaked in during the idle period. In larger equipment, constancy of gage readings may be acceptable instead of the "no bubble" method.

4. Leaks may also be detected by putting a **refrigerant** or **air pressure** of at least 50 psi in the equipment, and then spreading a thick, ropey soap solution or heavy, clear oil over joints or other suspected places. Presence of bubbles indicates and locates the leaks.

5. Another method is to put the equipment under a pressure of 50 psi or more of one of the halogen refrigerants, and then test each joint or suspected place with the exploring tube of a **"halide torch"** burning either methyl alcohol or Prestolite gas. This is a very accurate method which is becoming increasingly popular.

If desired, the pressure of the refrigerant may be built up by adding with it carbon dioxide or nitrogen to a pressure high enough to accentuate small leaks. If this method is used with methyl chloride or other flammable halogen, due caution must be exercised to provide sufficient ventilation to prevent ignition of the refrigerant by the flame of the torch.

6. The **"smoke"** method is usable with ammonia and sulfur dioxide. If a swab dipped into "stronger" ammonia (about 25% solution of aqueous ammonia) is brought near a leak from which sulfur dioxide is emerging, a white smoke is easily visible, indicating the presence and relative location of the leak.

Conversely, ammonia equipment may be tested for leaks by burning a sulfur candle (thus producing sulfur dioxide) and exploring all suspected places, a leak evidencing itself by the "smoke." If the leak is large enough to pollute the nearby air considerably, its exact location may be difficult to determine by this method.

To detect or locate very small ammonia leaks, a piece of porous paper soaked in a phenolphthalein solution or Nessler solution, is applied to the suspected joint. Appearance of a red spot in the paper indicates the presence of a leak.

For practical purposes of maintenance cost and safety, a refrigerant should have some simple, positive method of locating leaks. The soap-and-water method can be used with any refrigerant but it is tedious, time-consuming and thus expensive. The "smoke" test method used with ammonia and sulfur dioxide is excellent.

The introduction of the halide torch has greatly simplified the location of leaks of the halogens. Its use makes it possible to detect and locate minute leaks of these refrigerants quickly and effectively.

IV. Action on Materials of which the Equipment is Constructed

The hydrocarbons attack natural **rubber,** causing it to become soft and to swell and to form blisters. For this reason the synthetic rubbers of the chloroprene or neoprene types, which are less vulnerable, are used.

In hermetically sealed systems the refrigerant comes into direct contact with the motor winding. The hydrocarbons are very effective solvents for many of the **electrical insulating varnishes** or **binders** with which the insulating fabrics are impregnated. Much trouble has been experienced from the refrigerant washing out these varnishes and depositing them, sometimes with fragments of the fabrics, on valve seats and in valve mechanisms or small orifices. Insulations of bakelite or glass fibre have been successfully used in such applications.

Moisture and/or air in the system seem to aggravate any action that the anhydrous refrigerant alone might have on metals, fibrous materials, gaskets, oils or other substances which may be exposed to the refrigerant. Ammonia readily unites with water to form aqueous ammonia, a strong alkaline solution that attacks **copper, brass** or other cupreous alloys.

Inasmuch as in ordinary practice there is some moisture in the system, copper or copper alloys are not used in ammonia systems where the ammonia may come into contact with them.

Another well known instance of a refrigerant attacking materials of which the system is made is the case of sulfur dioxide that has been exposed to moisture. Sulfurous acid is formed, which strongly attacks the metals in the system, especially the **ferrous metals.** Depending upon the amount of moisture present, the resulting corrosion may range from mere discoloration of the metals up to "freezing" of pistons in cylinders, shafts in bearings or other damage, even to the extent of being entirely irreparable.

If free **aluminum, magnesium** or **zinc** or their alloys or die-castings containing them, come into contact with methyl chloride, they are strongly attacked. (There is no action of methyl chloride on activated alumina, the drying agent.) The resulting compounds react with the oil to form a heavy black gummy precipitate that stops up screens and oil passages and fouls valves, rings, bearings, and other working parts.

The action of methyl chloride on aluminum produces aluminum tri-methyl, a highly inflammable gas **that may ignite with explosive violence** when merely exposed to the air or abnormally high pressures. Aluminum or its alloys must not be used with methyl chloride, and if such a system is found, **much caution must be used** in opening it for repair.

Methyl chloride and the Freons attack magnesium, and in additon methyl chloride attacks zinc. (Zinc is sometimes used in dehydrators with sulfur dioxide as a "neutralizer" for sulfurous acid.) Magnesium and zinc are corroded by such action, but there are no known cases of formation of dangerous compounds. To some degree all of the halogenated hydrocarbons, which include methyl chloride, ethyl chloride, the Freons and methylene chloride, unite with water even in small quantities to form acids, particularly hydrochloric acid, which will readily attack any of the metals commonly used in refrigerating equipment, as well as some of the materials of which gaskets, electrical insulation, and other non-metallic materials may be composed.

Copper plating. In systems employing methyl chloride, some of the Freons or other halogenated hydrocarbon refrigerants, certain types of oils have elements in them that dissolve copper from tubing or other parts, as the oil is being circulated throughout the system. The copper thus picked up is deposited on hot surfaces such as bearings, shafts, bushings, pins, pistons, rings, cylinders, valves and valve seats. There the copper accumulates, in time causing seizure of frictional parts or leakage of valves.

Moisture in the system aggravates this action but is not its direct cause. The use of a more highly refined oil with a small content of unsaturated hydrocarbons, greater stability and low Sligh oxidation number, plus thoroughly drying the system, will practically eliminate instances of "copper plating."

All of the statements above emphasize the necessity of properly drying the system and keeping it dry, and of introducing into the system only clean, dry refrigerant and suitable oil.

The straight hydrocarbons such as ethane, propane, butane, etc., containing carbon and hydrogen only, are less susceptible to formation of harmful compounds in the presence of moisture and air than the halogens that include fluorine or chlorine atoms. Therefore moisture in systems using the straight hydrocarbons may present a less serious problem, except that trouble will continue to result from freeze-ups at the expansion valve.

V. Safety Considerations (Table 4)

In choosing a refrigerant for a specific application, safety must be considered as a relative term. A refrigerant such as ammonia might not, because of its explosive and toxic characteristics, be considered safe for an automatic, small commercial or household application, yet is quite acceptable, from a safety viewpoint, for a large cold storage or ice-making plant in which the equipment is under constant surveillance by experienced operating personnel. The quantity of refrigerant used, the size of the room into which it might leak, whether or not flames are present, the type of occupancy and the odor of the refrigerant also have a direct bearing on the relative safety of the installation. A large multiple installation using methyl chloride would be unsafe in an institution, hotel or apartment house, while a small household refrigerator or freezer with only a pound or two of methyl chloride as the total charge, would be safe for all practical purposes.

Table 4. Relative Safety of Refrigerants

Refrigerant	ASA Safety Code group classification	Nat'l Fire Underwriters group number	Toxicity Lethal or serious injury[3]					Flammable or explosive limits of concentration in air
			Refrigerant in air			Products of decomposition by flame		
			Duration of exposure hours	Percentage by volume	Pounds per 1,000 cu ft	Duration of exposure min	Percentage by volume[4]	Percentage by volume
Methane	3[1]	+5						4.9–15.0
Ethylene	3[1]	+5						3.0–25.0
Nitrous oxide			8	.0025				Non-Flam.
Ethane	3	5	2	37.5–51.7				3.3–10.6
Carbon dioxide	1	5	½ to 1	29–30	33.2–34.3			Non-Flam.
Propane	3	5	2	37.5–51.7	42.4–58.5			2.3–7.3
Freon-22	1[1]	5A				16	1.0	Non-Flam.[2]
Ammonia	2	2	½	.5–.6	.221–.256			16.0–25.0
Carrene-7	1[1]	5A	2	19.4–20.3	50.2–52.2	25	1.1	Non-Flam.
Freon-12	1	6	2	28.5–30.4	89.6–95.7	20	1.0	Non-Flam.
Methyl chloride	2	4	2	2–2.5	2.62–3.28	30	2.4	8.1–17.2
Isobutane	3	+5						1.8–8.4
Sulfur dioxide	2	1	1/12	.7	1.165			Non-Flam.
Butane	3	5	2	37.5–51.7				1.6–6.5
Freon-114	1	6	2	20.1–21.5	90.5–96.8	15	1.0	Non-Flam.
Freon-21	1		½	10.2	27.1			Non-Flam.[3]
Ethyl chloride	2	4	1	4.0	6.72	18	2.0	3.7–12.0
Freon-11	1	5	2	10	35.7	5	1.0	Non-Flam.
Methyl formate	2	3	1	2–2.5	3.12–3.9			4.5–20.0
Methylene chloride	1	4A	½	5.1–5.3	11.25–11.7	20	1.0	Non-Flam.
Freon-113	1[1]	4	1	4.8–5.2	23.3–25.2	16	1.2	Non-Flam.
Dichlorethylene	2	4	1	2–2.5	5.04–6.3	5	2.1	5.6–11.4

[1] Unofficial.

[2] Very slightly flammable, but for practical purposes considered non-flammable.

[3] To guinea pigs.

[4] Initial concentration.

Safety may be considered with respect to toxicity, inflammability and explosiveness.

1. **Toxicity.** In a strict sense, all of the refrigerants except air, which is rarely used, may cause suffocation by oxygen deficiency.

The term "toxic" is usually applied to those refrigerants which are actually injurious to human beings—those that may cause death or serious injury even though they are mixed in air in small enough proportions that sufficient oxygen can be obtained from the air-refrigerant vapor mixture.

The extent and permanence of the injury depend upon: (1) the nature of the refrigerant, i.e., its relative toxicity; (2) the percentage of the refrigerant in the air; (3) the duration of time that the person is subjected to the mixture. Obviously, the greater the percentage of concentration of refrigerant in the air and the longer the person is subjected to the vapor, the greater will be the injury, or, in fatal instances, the sooner death will occur. The National Board of Fire Underwriters has tested most of the commonly used refrigerants and classified them into groups according to their toxicity. Group 1 includes the most toxic, with toxicity decreasing as the group number increases, so that Group 6 includes those refrigerants which are harmless, except when a high concentration might dangerously exclude oxygen from the person exposed.

In Table 4, the refrigerants are classified as to toxicity according to the Underwriters' tests.

Mixtures of toxic vapors and air may attack the membranes of the lungs, nose, eyes or even the skin, and/or they may be absorbed into the blood and carried to other parts of the body to which they may be injurious.

Some refrigerants which in their normal state mixed with air, even in heavy percentages, are practically harmless, are decomposed by contact with a flame and form products that are highly toxic and even in low concentrations may cause fatal or serious injury in less than one-half hour. The halogens, including methyl chloride and the Freons, are in this category. If a flame or an electric heating element is present in a room into which these refrigerants are known or suspected of having leaked, the room should be well ventilated before entering.

2. **Flammability and explosiveness.** Many of the refrigerants that have been in common and widespread use for many years will burn if mixed with just the right percentages of air, and yet fires or explosions from this cause have been extremely rare. This hazard is slight because: (1) the flammable vapor-air mixture must be within a rather narrow limit of concentration of the vapor; (2) the refrigerant vapor is usually under sufficient pressure that it tends to blow out the flame; and (3) due to offensive odor or lack of refrigeration, the loss of refrigerant is usually detected before an inflammable concentration develops.

In Table 4, the various refrigerants that have been or are in fairly common use are classified as to flammability. Some refrigerants, such as most of the Freons (those from which the hydrogen atom has been eliminated), will not burn nor will they support combustion. Others will burn, but only feebly and within close limits of concentration of air. Ammonia and methyl chloride fall in this second group. The third group includes those that are quite flammable, such as methane, ethylene, ethane, propylene, propane, butane, and isobutane.

In general, those halogenated hydrocarbons, such as Freons-11, 12, 113 and 114, in which the hydrogen atoms have been entirely replaced by fluorine and/or chlorine atoms, are the least toxic and the least inflammable. These hydrocarbons in which the hydrogen atoms have not been entirely removed are moderately toxic and slightly flammable. The straight hydrocarbons are the most toxic and the most flammable (see bibliography, page 207).

The ASRE-sponsored ASA "American Standard Safety Code for Mechanical Refrigeration" is usually the basic model on which states and municipalities compile their codes. Despite this fact, there is much variance between the provisions of the various local codes. Since these refrigeration codes are units of local building or safety codes, they must be taken into consideration and followed when installing, repairing or operating refrigeration or air-conditioning equipment in any area under the jurisdiction of a local code.

VI. Cost

Other factors being equal, low cost of a refrigerant is quite desirable. However, the price per pound must be evaluated on the basis of: (1) the number of pounds that must be circulated per minute to provide the necessary refrigeration (dependent upon the "net refrigerating effect" per pound of refrigerant); (2) the density of the liquid. Since refrigerants are sold by the pound rather than by unit of volume, the cost advantage is with those having high net refrigerating effects and low liquid densities. Such refrigerants also permit smaller and less costly liquid receivers and piping.

VII. Type of Compressor

In late years, advancements in metallurgy have made new alloys available and modern precision machining processes have allowed finer finishes, so that much greater running speeds than heretofore are entirely practical This has resulted in compressors driven directly by induction electric motors at full motor speed. Reduction of size, further aided by a multiplicity of cylinders, has permitted cost reduction and has stimulated the development and use of self-contained and portable fixtures. These advances in compressor design have affected the requirements of the refrigerant, putting increased emphasis on horsepower per ton, amount of refrigerant circulated, vapor density, viscosity and electrical resistance, and reducing the importance of compression ratio and evaporator pressure.

VIII. Methods of Handling

During storage, transport, transfer from larger to smaller cylinders, and installations, service and maintenance operations, refrigerants must be handled by experienced workmen who are familiar with their nature and with the precautions that must be taken.

Reference should be made to instructions issued by the refrigerant manufacturers on the handling of the refrigerant that they produce, and on the safe quantities of refrigerant that may be put into their particular containers.

The Interstate Commerce Commission regulates the type, size, strength, test pressures, design, marking and retest period of refrigerant containers for interstate shipment of the various refrigerants. ICC regulations must be complied with when shipping and handling refrigerants.

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Calculated values based whenever possible on data in Chapter 7.

9. BRINE PROPERTIES

1. **Brine.** Tabular material on brines begins with the two common salts, calcium chloride and sodium chloride. The properties of pure calcium chloride brine and sodium chloride brine are given in **Tables 1** and **6**. For commercial grades use the formulae indicated in the footnotes to these tables. **Tables 2** and **7**, which give for these salts the enthalpy for liquid and frozen brines, may be used for computa-

Table 1. Properties of Pure Calcium Chloride Brine

Pure $CaCl_2$ % by weight	Specific gravity 60 F	Beaumé density 60 F	Deg sal-ometer 60/60 F	Specific heat 60 F Btu per lb F	Freezing point F	Weight per gallon*			Weight per cubic foot†		
						$CaCl_2$ lb/gal	Water lb/gal	Brine lb/gal	$CaCl_2$ lb/cu ft	Water lb/cu ft	Brine lb/cu ft
0	1.000	0.0	0.0	1.000	32.0	0.000	8.34	8.34	0.00	62.40	62.40
5	1.044	6.1	24.4	0.924	27.7	0.436	8.281	8.717	3.26	61.89	65.15
6	1.050	7.0	28.0	0.914	26.8	.526	8.234	8.760	3.93	61.59	65.52
7	1.060	8.2	32.8	0.898	25.9	.620	8.231	8.851	4.63	61.51	66.14
8	1.069	9.3	37.2	0.884	24.6	.714	8.212	8.926	5.34	61.36	66.70
9	1.078	10.4	41.6	0.869	23.5	.810	8.191	9.001	6.05	61.22	67.27
10	1.087	11.6	46.4	0.855	22.3	0.908	8.168	9.076	6.78	61.05	67.83
11	1.096	12.6	50.4	0.842	20.8	1.006	8.137	9.143	7.52	60.81	68.33
12	1.105	13.8	55.2	0.828	19.3	1.107	8.120	9.227	8.27	60.68	68.95
13	1.114	14.8	59.2	0.816	17.6	1.209	8.093	9.302	9.04	60.47	69.51
14	1.124	15.9	63.6	0.804	15.5	1.313	8.064	9.377	9.81	60.27	70.08
15	1.133	16.9	67.6	0.793	13.5	1.418	8.034	9.452	10.60	60.04	70.64
16	1.143	18.0	72.0	0.779	11.2	1.526	8.010	9.536	11.40	59.86	71.26
17	1.152	19.1	76.4	0.767	8.6	1.635	7.984	9.619	12.22	59.67	71.89
18	1.162	20.2	80.8	0.756	5.9	1.747	7.956	9.703	13.05	59.46	72.51
19	1.172	21.3	85.2	0.746	2.8	1.859	7.927	9.786	13.90	59.23	73.13
20	1.182	22.1	88.4	0.737	− 0.4	1.970	7.883	9.853	14.73	58.90	73.63
21	1.192	23.0	92.0	0.729	− 3.9	2.085	7.843	9.928	15.58	58.61	74.19
22	1.202	24.4	97.6	0.716	− 7.8	2.208	7.829	10.037	16.50	58.50	75.00
23	1.212	25.5	102.0	0.707	−11.9	2.328	7.792	10.120	17.40	58.23	75.63
24	1.223	26.4	105.6	0.697	−16.2	2.451	7.761	10.212	18.32	58.00	76.32
25	1.233	27.4	109.6	0.689	−21.0	2.574	7.721	10.295	19.24	57.70	76.94
26	1.244	28.3	113.2	0.682	−25.8	2.699	7.680	10.379	20.17	57.39	77.56
27	1.254	29.3	117.2	0.673	−31.2	2.827	7.644	10.471	21.13	57.12	78.25
28	1.265	30.4	121.6	0.665	−37.8	2.958	7.605	10.563	22.10	56.84	78.94
29	1.276	31.4	125.6	0.658	−49.4	3.090	7.565	10.655	23.09	56.53	79.62
29.6	1.290	32.6	—	0.655	−59.8	3.16	7.59	10.75	23.65	56.80	80.45
30	1.295	33.0	—	0.653	−50.8	3.22	7.58	10.80	24.06	56.70	80.76
32	1.317	34.9	—	0.640	−19.5	3.49	7.49	10.98	26.10	56.04	82.14
34	1.340	36.8	—	0.630	+ 4.3	3.77	7.40	11.17	28.22	55.35	83.57

Deg Beaumé heavy $= \dfrac{\text{deg } CaCl_2 \text{ sal.}}{4}$ * Wt of flake $CaCl_2 = \dfrac{\text{wt of pure } CaCl_2}{.785}$ † Wt of solid $CaCl_2 = \dfrac{\text{wt of pure } CaCl_2}{.74}$.

Table 2. Enthalpy ($CaCl_2$)

Enthalpy (Btu per lb from 32 F where it is 200 minus heat of solution)[1]

Sp gr		1.08	1.12	1.16	1.20	1.24	1.28
Lb $CaCl_2$ per 100 lb solution		9.2	13.5	17.6	21.5	25.1	28.7
Freezing point, deg F		23.2	16.5	7.0	−5.8	−21.5	−44.3
Lb solution per gal, 60 F		9.01	9.35	9.68	10.01	10.35	10.68
t, deg F							
−65	Solid	41.14	54.70	67.69	80.22	92.27	103.80
−60		43.07	56.59	69.54	82.01	94.02	105.52
−59.8		43.15	56.65	69.61	82.08	94.08	105.58
−59.8	Liquid and Solid	60.60	82.30	103.13	123.16	142.43	160.93
−55		63.34	85.40	106.57	126.93	146.52	165.30
−50		66.25	88.67	110.23	130.94	150.87	169.97
−45		69.31	92.20	114.18	135.30	155.64	175.13
−40		72.57	95.96	118.44	140.04	160.82	180.24
−35		76.01	100.02	123.08	145.25	166.58	183.41
−30		79.71	104.43	128.15	150.96	172.90	186.59
−25		83.72	109.24	133.77	157.34	180.01	189.77
−20		88.07	144.63	140.13	164.64	187.57	192.95
−15		93.01	120.76	147.41	173.04	190.87	196.15
−10		98.64	127.91	156.04	183.06	194.19	199.35
−5		105.07	136.36	166.42	192.45	197.51	202.55
0		112.60	146.29	178.67	195.95	200.84	205.77
5		121.63	158.44	193.87	199.45	204.18	208.99
10		133.07	174.13	198.38	202.97	207.53	212.21
15		149.39	197.04	202.11	206.49	210.89	215.45
20		176.58	201.63	205.86	210.03	214.27	218.70
25	Liquid	210.49	205.63	209.62	213.58	217.75	221.94
30		205.79	209.65	213.38	217.14	221.04	225.20
35		210.12	213.67	217.15	220.70	224.44	228.46
40		214.44	217.70	220.94	224.28	227.85	231.74
45		218.78	221.74	224.74	227.87	231.27	235.02
50		223.11	225.79	228.54	231.47	234.69	238.31
55		227.46	229.85	232.35	235.08	238.13	241.62
60		231.82	233.91	236.17	238.70	241.58	244.93
65		236.18	237.98	240.00	242.32	245.03	248.24
70		240.55	242.07	243.83	245.95	248.50	251.57
75		244.92	246.17	247.67	249.59	251.97	254.86

Table 3. Calcium Chloride Density

Specific gravity refers to water at 60 F

% Salt	Density (gr per cc)			
	−4 F	14 F	32 F	50 F
5			1.043	1.042
6			1.052	1.051
7			1.061	1.060
8			1.071	1.069
9			1.080	1.078
10			1.089	1.087
11			1.098	1.096
12			1.108	1.105
13			1.117	1.115
14			1.127	1.124
15		1.139	1.137	1.134
16		1.149	1.146	1.143
17		1.159	1.156	1.153
18		1.169	1.166	1.163
19		1.180	1.176	1.173
20		1.190	1.186	1.183
22	1.215	1.211	1.207	1.203
24	1.236	1.232	1.228	1.224

tion of heat loads with ordinary brine as well as for any problems dealing with frozen brines.[1] **Tables 3** and **8** show the density, commonly used as the measure of salt concentration. Viscosities are given in **Tables 4** and **9.** Chapter 6, on fluid flow, presents additional data on viscosity. **Tables 15** and **16** show the thermal conductivity of calcium and sodium brines at varying temperatures and concentrations.[12]

Tables 5 and **10** give data on the vapor pressures existing over salt solutions, in equilibrium with them. The temperature corresponding to the vapor pressure given, can be considered the boiling point of the solution at that pressure. A common example of the use of Tables 5 and 10 is to determine the amount of water to be taken up by a salt in coming into equilibrium with the vapor pressure of the moisture existing in the atmosphere. These tables also show what temperature would result from the establishment of a stated pressure above a solution, a principle used in some refrigerating machines to establish temperatures below the ordinary.

An elementary method of producing a lowering of temperature is by the mixing of salts. The effect of several mixtures is indicated in **Table 11.**

2. Freezing of brines. A solution of any salt in water has a certain concentration at which the freezing point is lowest. A solution of this concentration is called a **eutectic mixture,** or **cryohydrate,** and the temperature at which it freezes is the eutectic temperature. A solution at any other concentration will start to freeze at a higher temperature, and as the temperature is lowered, will freeze out pure ice or salt, the remaining liquid approaching the eutectic concentration, until the final freezing takes place at the eutectic point.

Referring to **Fig. 1,** if a solution of concentration x, less than the eutectic, at a temperature above 32 F, is cooled, it will not solidify when 32 is reached (point A), but will continue to cool as a liquid until point B is reached. At this point ice crystals of pure water will begin to form, accompanied by the removal of their latent heat. The removal of this water from the solution increases the concentration of the residual solution. As the temperature is lowered these crystals continue to form, and the mixture of ice crystals and brine solution forms a slush. When point C is reached, there is a mixture of ice crystals, C_2, and brine solution of concentration C_1, in the proportions of l_1 parts of brine to l_2 parts of ice crystals in (l_1+l_2) parts of mixture. When the process has continued to

Table 4. Calcium Chloride Viscosity

%	Viscosity (in centipoises)[8]				
	−13 F	−4 F	14 F	32 F	50 F
1.6				1.804	1.319
3.4				1.854	1.358
5.0				1.922	1.407
6.8				2.001	1.461
8.7				2.104	1.515
10.5				2.211	1.608
12.2				2.334	1.702
14.1				2.491	1.809
16.0			4.217	2.677	1.947
17.8			4.521	2.888	2.113
19.8			4.894	3.119	2.310
21.8		8.522	5.335	3.413	2.560
23.7	11.499	9.326	5.850	3.761	2.844
25.7	12.901	10.513	6.585	4.217	3.197
27.6	14.955	12.013	7.571	4.854	3.678
29.7		14.269	8.860	5.688	4.335
34.0				8.164	6.119
37.2				10.861	7.870

Table 5. Calcium Chloride Vapor Pressure

Vapor pressure above solutions, mm Hg[a]

Sp gr at 60 F	Temperature, deg F													
	42.8	39.2	35.6	32	28.4	24.8	21.2	17.6	14	10.4	6.8	3.2	−.4	−4.0
1.00	7.01	6.10	5.29	4.58	3.88	3.28	2.76	2.32	1.95	1.63	1.36	1.13	0.94	0.77
1.01	6.97	6.07	5.26	4.56										
1.02	6.93	6.03	5.23	4.53										
1.03	6.90	6.00	5.20	4.50										
1.04	6.86	5.96	5.17	4.48	3.86									
1.05	6.81	5.92	5.13	4.45	3.83									
1.06	6.76	5.88	5.10	4.42	3.81									
1.07	6.71	5.84	5.06	4.38	3.78	3.26								
1.08	6.66	5.79	5.02	4.35	3.75	3.23								
1.09	6.60	5.74	4.98	4.31	3.72	3.21	2.76							
1.10	6.56	5.70	4.95	4.28	3.69	3.18	2.74							
1.11	6.49	5.65	4.90	4.24	3.66	3.15	2.71	2.32						
1.12	6.41	5.58	4.84	4.19	3.61	3.11	2.68	2.29						
1.13	6.36	5.53	4.80	4.15	3.58	3.09	2.65	2.27	1.94					
1.14	6.28	5.46	4.74	4.10	3.54	3.05	2.62	2.25	1.92					
1.15	6.20	5.40	4.68	4.05	3.49	3.01	2.59	2.22	1.90	1.62				
1.16	6.13	5.33	4.62	4.00	3.45	2.97	2.56	2.19	1.87	1.60	1.35			
1.17	6.03	5.25	4.55	3.94	3.40	2.93	2.52	2.16	1.84	1.57	1.33			
1.18	5.93	5.16	4.47	3.87	3.34	2.88	2.48	2.12	1.81	1.54	1.31	1.11		
1.19	5.83	5.07	4.40	3.81	3.29	2.83	2.44	2.09	1.78	1.52	1.29	1.09	0.92	
1.20	5.71	4.97	4.31	3.73	3.22	2.77	2.39	2.04	1.75	1.49	1.26	1.07	0.91	0.76
1.21	5.59	4.87	4.22	3.65	3.15	2.72	2.34	2.00	1.71	1.46	1.24	1.05	0.89	0.75
1.22	5.47	4.76	4.12	3.57	3.08	2.65	2.28	1.95	1.67	1.42	1.21	1.03	0.87	0.73
1.23	5.30	4.61	4.00	3.46	2.99	2.57	2.21	1.90	1.62	1.38	1.17	1.00	0.85	0.71
1.24	5.15	4.48	3.88	3.36	2.90	2.50	2.15	1.84	1.57	1.34	1.14	0.97	0.82	0.69
1.25	4.98	4.33	3.75	3.25	2.80	2.42	2.08	1.78	1.52	1.29	1.10	0.93	0.79	0.66
1.26	4.77	4.14	3.59	3.11	2.68	2.31	1.99	1.70	1.46	1.24	1.05	0.89	0.75	0.63

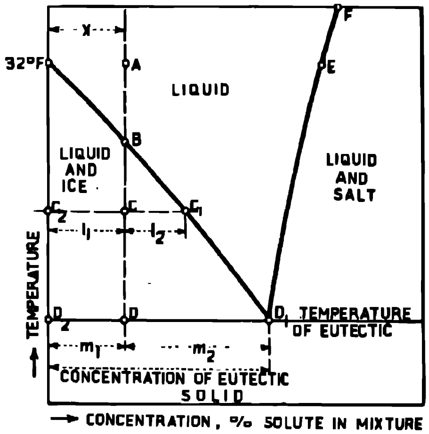

Fig. 1. Brine Mixture

point D, there is a mixture of m_1 parts of eutectic brine solution D_1, and m_2 parts of ice, D_2, all at the eutectic temperature. As more heat is removed, the m_1 parts of eutectic brine freeze at uniform temperature, until all of the latent heat is removed. The frozen eutectic mixture is a mechanical mixture of salt and frozen water, not a solution, and consequently the latent heat must be corrected for the heat of solution. If this is positive it decreases the effective latent heat, if negative it increases it.

If the initial solution is of greater concentration than the eutectic, salt freezes out as the temperature is lowered, instead of water, and the concentration is decreased until at the eutectic temperature the eutectic concentration is reached. In brines used as a refrigerating fluid, salt sometimes freezes out due to the use of too high concentrations.

The upper right branch of the curve in Fig. 1 gives the solubility—point *E* at 32 F, point *F* at some higher temperature.

On melting a eutectic brine, the water must redissolve the salt, which may require agitation, circulation or other mixing aids.

The total heat of $CaCl_2$ solutions at various temperatures and concentrations is shown in Table 2. The -59.8 horizontal line represents the eutectic temperature, and the lower horizontal line in each column represents the temperature at which ice crystals start to freeze out. In Table 7 similar data are given for NaCl. **Table 12** gives the eutectic concentrations and temperatures for a number of two-component systems. The data are adapted from the International Critical Tables (which contain data on many additional systems, including some three-component systems).

Table 13 shows the concentration and freezing point of some aqueous solutions (non-eutectic). Other non-eutectic solutions can be found in the International Critical Tables.

Figs. 2 and **3** show the variations in specific heat of calcium chloride and sodium chloride brine solutions with changes in concentration and temperature.

The **applications** of brine in refrigeration are mainly in the industrial machinery field where the calcium chloride solution is the standard secondary refrigerating medium. Corrosion is the principal problem met in use of calcium chloride brines, es-

Table 6. Properties of Pure Sodium Chloride (NaCl) Brine

Pure NaCl % by wt	Specific gravity 59 F 39 F	Beaumé density 60 F	Salometer 59 F 39 F	Specific heat 59 F Btu/lb deg F	Freezing point deg F	Weight per gallon*			Weight per cubic foot*		
						NaCl lb/gal	Water lb/gal	Brine lb/gal	NaCl lb/cu ft	Water lb/cu ft	Brine lb/cu ft
0	1.000	0.0	0.0	1.000	32.0	0.000	8.34	8.34	0.000	62.40	62.4
5	1.035	5.1	18.2	0.938	27.0	0.432	8.22	8.65	3.230	61.37	64.6
6	1.043	6.1	22.5	0.927	25.5	0.523	8.19	8.71	3.906	61.19	65.1
7	1.050	7.0	26.0	0.917	24.0	0.613	8.15	8.76	4.585	60.91	65.5
8	1.057	8.0	29.6	0.907	23.2	0.706	8.11	8.82	5.280	60.72	66.0
9	1.065	9.0	33.5	0.897	21.8	0.800	8.09	8.89	5.985	60.51	66.5
10	1.072	10.1	37.2	0.888	20.4	0.895	8.05	8.95	6.690	60.21	66.9
11	1.080	10.8	41.1	0.879	18.5	0.992	8.03	9.02	7.414	59.99	67.4
12	1.087	11.8	44.8	0.870	17.2	1.090	7.99	9.08	8.136	59.66	67.8
13	1.095	12.7	48.7	0.862	15.5	1.188	7.95	9.14	8.879	59.42	68.3
14	1.103	13.6	52.6	0.854	13.9	1.291	7.93	9.22	9.632	59.17	68.8
15	1.111	14.5	56.8	0.847	12.0	1.392	7.89	9.28	10.395	58.90	69.3
16	1.118	15.4	60.0	0.840	10.2	1.493	7.84	9.33	11.168	58.63	69.8
17	1.126	16.3	64.0	0.833	8.2	1.598	7.80	9.40	11.951	58.36	70.3
18	1.134	17.2	68.0	0.826	6.1	1.705	7.76	9.47	12.744	58.06	70.8
19	1.142	18.1	71.7	0.819	4.0	1.813	7.73	9.54	13.547	57.75	71.3
20	1.150	19.0	75.2	0.813	+ 1.8	1.920	7.68	9.60	14.360	57.44	71.8
21	1.158	19.9	79.1	0.807	− 0.8	2.031	7.64	9.67	15.183	57.12	72.3
22	1.166	20.8	82.8	0.802	− 3.0	2.143	7.60	9.74	16.016	56.78	72.8
23	1.175	21.7	86.8	0.796	− 6.0	2.256	7.55	9.81	16.854	56.45	73.3
24	1.183	22.5	90.2	0.791	+ 3.8	2.371	7.51	9.88	17.712	56.09	73.8
25	1.191	23.4	94.0	0.786	+16.1	2.488	7.46	9.95	18.575	55.72	74.3
25.2	1.200				+32.0						

* $\text{Wt of commercial NaCl required} = \dfrac{\text{wt of pure NaCl required}}{\text{per cent purity}}$.

Table 7. Enthalpy (NaCl)

Enthalpy in Btu per lb from 32 F, where it is 200 minus heat of solution[1]

Sp gr		1.04	1.06	1.08	1.10	1.12	1.14	1.16	1.18
Lb NaCl per 100 lb solution		5.5	8.2	10.9	13.5	16.1	18.6	21.1	23.5
Freezing point, deg F		26.0	12.7	19.0	14.9	10.4	5.4	−0.3	−3.6
Lb NaCl per gal solution, 60 F		0.48	0.73	0.98	1.24	1.50	1.77	2.04	2.31
Lb solution per gal, 60 F		8.67	8.84	9.00	9.17	9.34	9.50	9.67	9.84
Deg F −6.03	Solid	46.46	50.34	54.35	58.29	62.17	65.96	69.98	72.93
−6.03	Liquid and Solid	70.26	86.12	101.77	117.16	132.29	147.11	161.65	174.35
−4		72.42	88.88	105.12	121.09	136.79	152.17	167.26	176.03
−2		74.65	91.74	108.60	125.19	141.48	157.45	173.13	177.59
0		77.05	94.86	112.43	129.71	146.70	163.34	178.98	179.16
2		79.66	98.28	116.67	134.74	152.49	169.90	180.56	180.73
4		82.50	102.06	121.35	140.33	158.98	177.25	182.15	182.29
6		85.61	106.22	126.57	146.57	166.23	183.47	183.74	183.86
8		89.09	110.95	132.50	153.73	174.56	185.09	185.33	185.43
10		93.10	116.45	139.48	162.16	184.41	186.71	186.92	187.00
12		97.72	122.90	147.70	172.13	188.00	188.33	188.51	188.58
14		103.14	130.47	157.44	183.96	189.66	189.96	190.11	190.15
16		109.60	139.62	169.24	190.85	191.31	191.58	191.71	191.72
18		117.62	151.09	184.13	192.54	192.57	193.21	193.30	193.30
20		127.97	166.04	193.64	194.24	194.63	194.83	194.90	194.88
22		141.96	186.43	195.39	195.94	196.29	196.46	196.50	196.45
24		162.72	196.38	197.14	197.63	197.94	198.09	198.10	198.03
26		196.19	198.18	198.88	199.33	199.61	199.72	199.70	199.61
28	Liquid	199.00	199.98	200.63	201.03	201.27	201.34	201.30	201.19
30		200.85	201.77	202.38	202.74	202.94	202.98	202.91	202.77
32		202.71	203.57	204.12	204.44	204.60	204.61	204.51	204.35
34		204.57	205.37	205.87	206.14	206.26	206.24	206.11	205.93
36		206.43	207.17	207.62	207.85	207.93	207.88	207.72	207.51
38		208.29	208.97	209.37	209.56	209.60	209.51	209.33	209.10
40		210.15	210.78	211.12	211.26	211.27	211.15	210.94	210.68
42		212.01	212.57	212.87	212.97	212.94	212.79	212.54	212.27
44		213.87	214.38	214.62	214.68	214.61	214.43	214.16	213.86
46		215.73	216.18	216.38	216.39	216.29	216.07	215.77	215.44
48		217.59	217.99	218.14	218.11	217.96	217.71	217.38	217.03
50		219.46	219.80	219.89	219.82	219.63	219.34	218.99	218.62

pecially in ice-making tanks where galvanized iron cans are immersed.

Calcium chloride is also used in machinery of intermediate sizes to provide holdover capacity in refrigeration storage tanks or plates. Corrosion treatment in this class of work is not readily applied by the user, but manufacturers may supply the treatment or service.

Ordinary salt is used where contact with calcium chloride could not be tolerated, as for example in the brine fog method of freezing fish and other foods. A common application now is in air cooling in unit coolers, as spray, preventing formation of frost on coils. For most refrigerating work the lower freezing point of calcium makes its use more convenient.

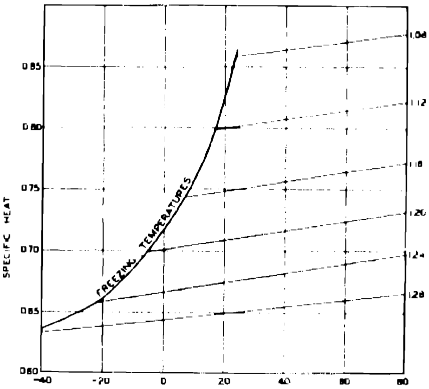

Fig. 2. Specific Heat of Calcium Chloride

Eutectic brine in metal containers is often used in place of dry ice for ice-cream cabinets and truck refrigeration. The eutectic brine, in metal containers, is frozen in cold rooms or by means of refrigeration coils built in the eutectic container.

Propylene glycol solution finds some use as a brine, principally in the brewing and ice cream industries. It is colorless, prac-

Table 8. Sodium Chloride Density

Specific gravity refers to water at 60 F

Sp gr	Densities (gr per cc)			
%	14 F	32 F	50 F	68 F
5		1.0382	1.0366	1.0341
6		1.0459	1.0440	1.0413
7		1.0536	1.0515	1.0486
8		1.0613	1.0590	1.0559
9		1.0691	1.0665	1.0633
10		1.0769	1.0741	1.0707
11		1.0849	1.0817	1.0782
12		1.0925	1.0897	1.0857
13		1.1004	1.0971	1.0933
14		1.1083	1.1048	1.1009
15	1.1195	1.1163	1.1126	1.1086
16	1.1277	1.1243	1.1205	1.1163
17	1.1359	1.1323	1.1284	1.1241
18	1.1442	1.1404	1.1363	1.1319
19	1.1535	1.1486	1.1444	1.1398
20	1.1608	1.1568	1.1524	1.1478
21	1.1692	1.1651	1.1606	1.1559
22	1.1777	1.1734	1.1688	1.1640
23	1.1862	1.1818	1.1771	1.1721
24	1.1948	1.1902	1.1854	1.1804

Fig. 3. Specific Heat of Sodium Chloride

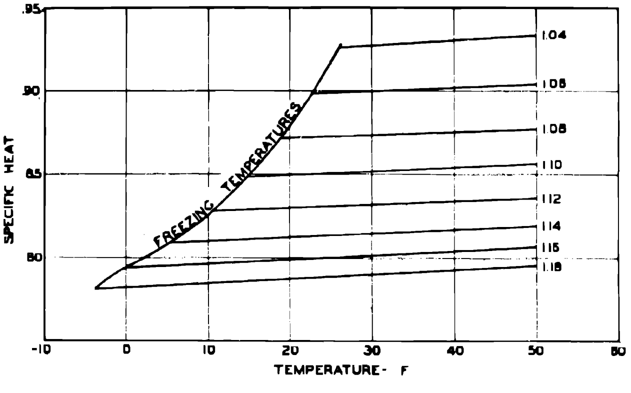

Table 9. Sodium Chloride Viscosity

	Viscosities (centipoises)[8]				
%	5 F	14 F	23 F	32 F	55 F
4.9				1.824	1.368
5.9				1.849	1.388
7.0				1.878	1.412
8.0			2.297	1.912	1.437
9.1			2.349	1.952	1.466
10.2			2.403	1.991	1.496
11.3			2.464	2.040	1.530
12.3			2.525	2.091	1.564
13.4		3.212	2.499	2.153	1.606
14.5		3.320	2.682	2.224	1.649
15.6		3.437	2.768	2.308	1.701
16.7		3.575	2.874	2.391	1.755
17.9	4.629	3.741	3.011	2.490	1.813
19.0	4.830	3.908	3.148	2.592	1.872
20.1	5.056	4.009	3.295	2.702	1.941
21.2	5.291	4.310	3.454	2.820	2.004
22.3	5.556	4.521	3.624	2.947	2.098
23.5	5.865	4.791	3.817	3.091	2.199
24.7		5.105	4.026	3.246	2.316
25.9				3.433	2.463

tically odorless, non-corrosive chemically and inhibits bacterial growth. **Table 14** gives the properties of pure propylene glycol. In solution the specific gravity varies little, being 1.020 at 20% glycol and 1.055 at 60% glycol, both at 23 F.

For information on the application of these brines in respect to brine coolers, refer to Chap. 26.

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Table 10. Sodium Chloride Vapor Pressure[8]

Vapor pressure above solutions in mm of mercury[8]

Sp gr at 60 F	Temperature, deg F													
	42.8	39.2	35.6	32.0	28.4	24.8	21.2	17.6	14.0	10.4	6.8	3.2	−0.4	−4.0
1.00	7.01	6.10	5.29	4.58	3.88	3.28	2.76	2.32	1.95	1.63	1.36	1.13	0.94	0.77
1.01	6.94	6.04	5.24	4.54										
1.02	6.88	5.99	5.20	4.50										
1.03	6.82	5.94	5.15	4.46	3.84									
1.04	6.76	5.89	5.10	4.42	3.81									
1.05	6.69	5.82	5.05	4.37	3.77	3.25								
1.06	6.62	5.76	5.00	4.33	3.74	3.22								
1.07	6.56	5.70	4.95	4.28	3.70	3.18								
1.08	6.48	5.64	4.89	4.23	3.65	3.15	2.71							
1.09	6.41	5.58	4.83	4.19	3.61	3.11	2.68	2.29						
1.10	6.34	5.52	4.78	4.14	3.57	3.08	2.65	2.27	1.94					
1.11	6.26	5.45	4.72	4.09	3.54	3.04	2.61	2.24	1.91					
1.12	6.17	5.37	4.66	4.03	3.48	3.00	2.58	2.21	1.89	1.61				
1.13	6.09	5.30	4.60	3.98	3.48	2.96	2.54	2.18	1.86	1.59	1.35			
1.14	6.00	5.22	4.53	3.92	3.38	2.91	2.51	2.15	1.83	1.56	1.33	1.13		
1.15	5.91	5.14	4.46	3.86	3.33	2.78	2.47	2.11	1.81	1.54	1.31	1.11		
1.16	5.82	5.06	4.39	3.80	3.28	2.82	2.43	2.08	1.77	1.52	1.29	1.09	0.92	
1.17	5.17	4.97	4.30	3.73	3.22	2.77	2.38	2.04	1.74	1.49	1.26	1.07	0.90	0.76

Table 11. Endothermal Mixtures[9]

Mixtures		Temperature drop, deg F
Ammonium nitrate Water	1 1	From +40 to + 4
Ammonium chloride Potassium nitrate Water	5 5 16	From +50 to +10
Ammonium chloride Potassium nitrate Sodium sulfate Water	5 5 8 16	From +50 to + 4
Sodium nitrate Nitric acid, diluted	3 2	From +50 to − 3
Ammonium nitrate Sodium carbonate Water	1 1 1	From +50 to − 7
Sodium phosphate Nitric acid, diluted	9 4	From +50 to −12
Sodium sulfate Sulfuric acid, diluted	5 4	From +50 to + 3
Sodium sulfate Ammonium chloride Potassium nitrate Nitric acid, diluted	6 4 2 4	From +50 to −10
Sodium sulfate Ammonium nitrate Nitric acid, diluted	6 5 4	From +50 to −40
Snow or pounded ice Sodium chloride	2 1	From +50 to − 5
Snow or pounded ice Sodium chloride Ammonium chloride	5 2 1	From +50 to −12
Snow or pounded ice Sodium chloride Ammonium chloride Potassium nitrate	24 10 5 5	From +50 to −18
Snow or pounded ice Sodium chloride Ammonium nitrate	12 5 5	From +50 to −25
Snow Sulfuric acid, diluted	3 2	From +32 to −23
Snow Hydrochloric acid	8 5	From +32 to −27
Snow Nitric acid, diluted	7 4	From +32 to −30
Snow Calcium chloride	4 5	From +32 to −40
Snow Calcium chloride, crystallized	2 3	From +32 to −50
Snow Potash	3 4	From +32 to −51

Table 12. Eutectic Points of Aqueous Solutions[5]

Substance	Eutectic concentration wt of solute ÷ wt of solution	Eutectic temperature, deg F
Ammonium nitrate (NH_4NO_2)	.428	+ 2.0
Ammonium chloride (NH_4Cl)	.191	+ 4.3
Calcium chloride ($CaCl_2$)	.324	− 59.8
Sodium chloride ($NaCl$)	.233	− 6.0
Sodium sulfate (Na_2SO_4)	.049	+ 30.0
Sodium nitrate ($NaNO_3$)	.397	+ 0.6
Sodium carbonate (Na_2CO_3)	.080	+ 28.2
Potassium chloride (KCl)	.197	+ 12.8
Potassium sulfate (K_2SO_4)	.065	+ 28.3
Potassium nitrate (KNO_3)	.112	+ 26.6
Potassium carbonate (KCO_3)	.355	− 34.7
Methyl alcohol (CH_4O)	.695	−164
Ethyl alcohol (C_2H_6O)	.763	− 94
Glycerine ($C_3H_6O_3$)	.669	− 60.4

Table 13. See page 218.

Table 14. Properties of Pure Propylene Glycol

Molecular weight	76.06
Boiling point, deg F	369.1
Specific gravity at 68 F	1.0381
Weight per gal at 68 F, lb	8.64
Surface tension at 77 F, dynes per cm	40.1
Viscosity at 77 F, poises	0.406
Specific heat at 68 F, Btu per lb F	0.590
Latent heat of evaporation at boiling point, Btu per lb	303.5
Flash Point (ASTM Open Cup), deg F	210.2
Coefficient of expansion at 68 F (per deg F)	0.000386

Table 13. Freezing Points of Aqueous Solutions

Alcohol		Glycerine		Ethylene Glycol		Propylene Glycol	
% by wt	deg F	% by wt	deg F	% by vol	deg F	% by vol	deg F
5	28.0	10	29.1	15	22.4	5	29.0
10	23.6	20	23.4	20	16.2	10	26.0
15	19.7	30	14.9	25	10.0	15	22.5
20	13.2	40	4.3	30	3.5	20	19.0
25	5.5	50	− 9.4	35	− 4.0	25	14.5
30	− 2.5	60	−30.5	40	−12.5	30	9.0
35	−13.2	70	−38.0	45	−22.0	35	2.5
40	−21.0	80	− 5.5	50	−32.5	40	− 5.5
45	−27.5	90	+29.1			45	−15.0
50	−34.0	100	+62.6			50	−25.5
55	−40.5					55	−39.5
						59	−57.0

Above 60% fails to crystallize at −99.4 F

Table 15. Thermal Conductivity of $CaCl_2$ Brine[12]

Btu ft per hr sq ft F

% salt	68 F	50 F	32 F	14 F	−4 F	−22 F	−40 F
5	0.343	0.333	0.324				
10	0.338	0.329	0.321				
15	0.335	0.326	0.317	0.308			
20	0.331	0.323	0.314	0.305			
25	0.328	0.319	0.310	0.302	0.294	0.286	
30	0.323	0.315	0.307	0.298	0.291	0.283	0.275
31.4	0.323	0.314	0.305	0.297	0.290	0.282	0.274 Eutectic
35	0.321	0.313	0.303	0.295			
40	0.317	0.309					

Table 16. Thermal Conductivity of NaCl Brine[12]

Btu ft per hr sq ft F

% salt	68 F	50 F	32 F	14 F	−4 F	−22 F	−40 F
0	0.346	0.337	0.327				
5	0.343	0.334	0.325				
10	0.339	0.331	0.322				
15	0.336	0.328	0.319	0.310			
20	0.334	0.325	0.316	0.307			
23.5	0.332	0.323	0.315	0.305	0.296	Eutectic at −6 F	
25	0.331	0.322	0.313	0.304			

10. PSYCHROMETRIC TABLES

TABLES presented here include data on the properties of dry air, saturated ice, saturated water, saturated steam, and mixtures of dry air and saturated steam in the range of temperatures from −160 F to 200 F.

Although properties of dry air and of the mixture of dry air and saturated steam are given, directly, only for a total pressure of 29.921 in. Hg abs, sufficient information is given so that these properties may be found for any other total pressure. **Table 3** shows for the range in question all of the items in the following development. **Tables 1 and 2** are arranged for general use with certain of the developmental items omitted.

There follows a summary of data used in preparation of the tables:

Assumed composition of dry air:

Constituent	*Fraction by volume*	*Fraction of weight*
Nitrogen	0.7803	0.7547
Oxygen	0.2099	0.2319
Monatomic gases (principally argon)	0.0094	0.0130
Carbon dioxide	0.0003	0.0004
Hydrogen	0.0001	0.0000

Based upon this analysis, the equivalent molecular weight of dry air is 28.966 lb per mol.

Equation of state for dry air is the Beattie-Bridgeman form with all virial coefficients beyond the second ignored.

$$v_a = v_1 - v_2$$

$$v_1 = \frac{53.35(t+459.7)}{p}$$

$$v_2 = -0.0225 + \frac{13.06}{t+459.7} + \frac{3.87(10)^6}{(t+459.7)^3}.$$

Specific heat at constant pressure of dry air. The values of the molal specific heat at zero pressure of nitrogen and oxygen as calculated from quantum theory and spectroscopic data by Johnston and Davis and Johnston and Walker were used to derive an empirical equation for the specific heat of dry air at zero pressure as a function of temperature. (For the monatomic gases present in dry air, a constant value of the specific heat at zero pressure of 0.124 Btu/lb F was used.)

For the range in temperature covered by the tables, the specific heat at zero pressure of dry air is:

$$c_p^{(0)} = 0.2397 + 4.08(10)^{-6}t + 9.24(10)^{-9}t^2$$

where t is in degrees F.

The enthalpy of dry air was calculated from the following equation:

$$h_a = h_1 - h_2,$$

where

$$h_1 = \int_0 c_p^{(0)}\,dt - h_{ao},$$

$$h_2 = \frac{B_a p_a}{778.18},$$

$$B_a = \frac{d\left(\frac{v_2}{T}\right)}{d\left(\frac{1}{T}\right)} = -.0225 + \frac{26.12}{(t+459.7)} + \frac{15.48(10)^6}{(t+459.7)^3}$$

The specific enthalpy of dry air was assigned, arbitrarily, a zero value at $t = 0$ F and at a dry air pressure of 29.921 in. Hg abs, so $h_{ao} = -0.1366$ Btu/lb.

Properties of saturated ice, saturated water, and saturated steam. The thermodynamic properties of ice, water, and steam were taken from the tables prepared by Goff and Gratch presented in *Heating, Piping and Air Conditioning*, February, 1946, and also in the 1946 A.S.H.V.E. Guide.

Properties of mixtures of dry air and saturated steam. In preparing these tables, the Gibbs-Dalton Law was assumed to

Table 1. Thermodynamic Properties of Air, Water and Steam (Continued)

Temperature, F	Properties of Water and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of water & steam, in. Hg	Enthalpy of sat water, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, ft^3	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_f	h_g	v_g	v_a	h_a	v_s	h_s	W_s
91	1.4671	59.0	1100.9	454.3	13.880	21.88	14.60	57.25	224.9
92	1.5136	60.0	1101.3	441.1	13.905	22.12	14.65	58.69	232.4
93	1.5613	61.0	1101.7	428.4	13.930	22.36	14.70	60.16	240.1
94	1.6103	62.0	1102.2	416.1	13.955	22.60	13.75	61.67	248.1
95	1.6607	63.0	1102.6	404.2	13.981	22.84	14.80	63.22	256.4
96	1.7124	64.0	1103.0	392.7	14.006	23.08	14.86	64.82	264.8
97	1.7655	65.0	1103.4	381.5	14.031	23.32	14.91	66.45	273.6
98	1.8200	66.0	1103.9	370.7	14.057	23.56	14.97	68.13	282.6
99	1.8759	67.0	1104.3	360.3	14.082	23.80	15.02	69.86	291.9
100	1.9334	68.0	1104.7	350.2	14.107	24.04	15.08	71.62	301.5
101	1.9923	69.0	1105.2	340.4	14.132	24.28	15.14	73.44	311.3
102	2.0529	70.0	1105.6	331.0	14.157	24.52	15.20	75.31	321.5
103	2.1149	71.0	1106.0	321.8	14.183	24.76	15.26	77.22	332.0
104	2.1786	72.0	1106.4	313.0	14.208	25.00	15.32	79.19	342.8
105	2.2440	73.0	1106.9	304.4	14.233	25.24	15.39	81.21	353.9
106	2.3110	74.0	1107.3	296.0	14.259	25.48	15.45	83.29	365.4
107	2.3798	75.0	1107.7	288.0	14.284	25.72	15.52	85.42	377.2
108	2.4503	76.0	1108.2	280.2	14.309	25.96	15.59	87.62	389.4
109	2.5226	77.0	1108.6	272.6	14.334	26.20	15.65	89.87	402.0
110	2.5968	78.0	1109.0	265.3	14.360	26.45	15.72	92.19	414.9
111	2.6728	79.0	1109.4	258.2	14.385	26.69	15.80	94.58	428.3
112	2.7507	80.0	1109.9	251.3	14.410	26.93	15.87	97.03	442.1
113	2.8306	81.0	1110.3	244.6	14.435	27.17	15.94	99.55	456.3
114	2.9125	82.0	1110.7	238.1	14.461	27.41	16.02	102.16	471.0
115	2.9963	83.0	1111.1	231.8	14.486	27.65	16.10	104.81	486.1
116	3.0823	84.0	1111.6	225.8	14.511	27.89	16.18	107.55	501.6
117	3.1703	85.0	1112.0	219.9	14.537	28.13	16.26	110.38	517.7
118	3.2606	86.0	1112.4	214.1	14.562	28.37	16.34	113.29	534.3
119	3.3530	87.0	1112.8	208.6	14.587	28.61	16.43	116.28	551.4
120	3.4477	88.0	1113.3	203.2	14.612	28.85	16.51	119.36	569.0
121	3.5446	89.0	1113.7	197.9	14.637	29.09	16.60	122.52	587.2
122	3.6439	90.0	1114.1	192.9	14.663	29.33	16.70	125.79	606.0
123	3.7455	91.0	1114.5	188.0	14.688	29.57	16.79	129.15	625.3
124	3.8496	92.0	1114.9	183.2	14.713	29.82	16.89	132.61	645.3
125	3.9561	93.0	1115.4	178.5	14.739	30.06	16.98	136.17	665.9
126	4.0651	94.0	1115.8	174.0	14.764	30.30	17.08	139.88	687.2
127	4.1768	95.0	1116.2	169.6	14.789	30.54	17.19	143.64	709.2
128	4.2910	96.0	1116.6	165.4	14.814	30.78	17.29	147.54	731.9
129	4.4078	97.0	1117.0	161.3	14.839	31.02	17.40	151.57	755.4
130	4.5274	98.0	1117.5	157.3	14.865	31.26	17.52	155.72	779.6
131	4.6498	99.0	1117.9	153.4	14.890	31.50	17.63	160.00	804.6
132	4.7750	100.0	1118.3	149.6	14.915	31.74	17.75	164.43	830.5
133	4.9030	101.0	1118.7	145.9	14.941	31.98	17.87	168.98	857.2
134	5.0340	102.0	1119.2	142.4	14.966	32.22	17.99	173.69	884.8
135	5.1679	103.0	1119.6	138.9	14.991	32.46	18.12	178.54	913.3
136	5.3049	104.0	1120.0	135.5	15.016	32.70	18.25	183.57	942.8
137	5.4450	105.0	1120.4	132.2	15.043	32.94	18.39	188.75	973.4
138	5.5881	106.0	1120.8	129.1	15.067	33.18	18.53	194.09	1005.0
139	5.7345	107.0	1121.2	126.0	15.092	33.43	18.67	199.64	1038
140	5.8842	108.0	1121.7	123.0	15.117	33.67	18.82	205.34	1071
141	6.0371	109.0	1122.1	120.0	15.143	33.91	18.97	211.27	1106
142	6.1934	110.0	1122.5	117.2	15.168	34.15	19.13	217.39	1143
143	6.3532	111.0	1122.9	114.4	15.193	34.39	19.29	223.70	1180
144	6.5164	112.0	1123.3	111.7	15.218	34.63	19.45	230.28	1219
145	6.6832	113.0	1123.7	109.1	15.244	34.87	19.62	236.94	1259
146	6.8536	114.0	1124.1	106.6	15.269	35.11	19.81	244.06	1301
147	7.0277	115.0	1124.6	104.1	15.294	35.35	19.99	251.34	1344
148	7.2056	116.0	1125.0	101.7	15.319	35.59	20.18	258.88	1389
149	7.3872	117.0	1125.4	99.32	15.345	35.83	20.37	266.71	1436
150	7.5727	118.0	1125.8	97.04	15.370	36.07	20.58	274.84	1485

Table 1. Thermodynamic Properties of Air, Water and Steam (Concluded)

Temperature, F	Properties of Water and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of water & steam, in. Hg	Enthalpy of sat water, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, ft³	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_f	h_g	v_g	v_a	h_a	v_g	h_s	W_s
151	7.7622	119.0	1126.2	94.81	15.395	36.31	20.79	283.25	1535
152	7.9556	120.0	1126.6	92.65	15.420	36.56	21.01	292.00	1587
153	8.1532	121.0	1127.0	90.54	15.446	36.80	21.23	301.07	1641
154	8.3548	122.0	1127.4	88.49	15.471	37.04	21.46	310.53	1698
155	8.5607	123.0	1127.8	86.50	15.496	37.28	21.71	320.34	1757
156	8.7708	124.0	1128.3	84.55	15.521	37.52	21.96	330.57	1818
157	8.9853	125.0	1128.7	82.66	15.547	37.76	22.22	341.18	1882
158	9.2042	126.0	1129.1	80.81	15.572	38.00	22.49	352.24	1948
159	9.4276	127.0	1129.5	79.02	15.597	38.24	22.77	363.29	2018
160	9.6556	128.0	1129.9	77.27	15.622	38.48	23.07	375.81	2090
161	9.8882	129.0	1130.3	75.56	15.648	38.72	23.37	388.34	2165
162	10.126	130.0	1130.7	73.90	15.673	38.96	23.69	401.45	2244
163	10.368	131.0	1131.1	72.28	15.698	39.21	24.02	415.14	2326
164	10.615	132.0	1131.5	70.71	15.724	39.45	24.37	429.44	2412
165	10.867	133.0	1131.9	69.17	15.749	39.69	24.73	444.41	2503
166	11.124	134.0	1132.3	67.67	15.774	39.93	25.11	460.10	2597
167	11.386	135.0	1132.7	66.21	15.799	40.17	25.50	476.53	2696
168	11.653	136.0	1133.1	64.79	15.824	40.41	25.92	493.77	2800
169	11.925	137.0	1133.5	63.40	15.850	40.65	26.35	511.83	2910
170	12.203	138.0	1133.9	62.04	15.875	40.89	26.81	530.86	3024
171	12.487	139.0	1134.4	60.73	15.900	41.13	27.29	550.89	3145
172	12.775	140.0	1134.8	59.44	15.925	41.37	27.79	571.97	3273
173	13.080	141.0	1135.2	58.18	15.951	41.61	28.32	594.19	3407
174	13.370	142.0	1135.6	56.96	15.976	41.85	28.88	617.65	3549
175	13.676	143.9	1136.0	55.77	16.001	41.10	29.47	642.48	3699
176	13.987	144.0	1136.4	54.60	16.026	42.34	30.09	668.67	3858
177	14.305	145.0	1136.8	53.47	16.052	42.58	30.75	696.49	4026
178	14.629	146.0	1137.2	52.36	16.077	42.82	31.46	726.04	4205
179	14.959	147.0	1137.6	51.28	16.103	43.06	32.21	757.46	4396
180	15.295	148.0	1137.9	50.22	16.128	43.30	32.99	790.88	4598
181	15.637	149.0	1138.3	49.19	16.153	43.54	33.83	826.46	4815
182	15.986	150.0	1138.7	48.19	16.178	43.78	34.74	864.74	5046
183	16.341	151.0	1139.1	47.20	16.203	44.02	35.70	905.58	5294
184	16.703	152.0	1139.5	46.25	16.229	44.26	36.74	949.49	5560
185	17.071	153.0	1139.9	45.31	16.254	44.51	37.85	996.65	5847
186	17.446	154.0	1140.3	44.40	16.279	44.75	39.04	1047.7	6156
187	17.829	155.0	1140.7	43.51	16.304	44.99	40.34	1102.8	6491
188	18.218	156.0	1141.1	42.64	16.329	45.23	41.75	1162.6	6854
189	18.614	157.0	1141.5	41.79	16.354	45.47	42.28	1227.8	7250
190	19.017	158.0	1141.9	40.96	16.380	45.71	44.94	1298.9	7682
191	19.428	159.1	1142.3	40.14	16.405	45.95	46.78	1376.9	8156
192	19.846	160.1	1142.7	39.35	16.430	46.19	48.79	1463.0	8679
193	20.271	161.1	1143.1	38.58	16.456	46.43	51.02	1558.2	9257
194	20.704	162.1	1143.5	37.82	15.481	46.68	53.50	1664.0	9901
195	21.145	163.1	1143.8	37.09	16.506	46.92	56.27	1782.6	10622
196	21.594	164.1	1144.2	36.36	16.531	47.16	59.40	1916.2	11434
197	22.050	165.1	1144.6	35.66	16.556	47.40	62.93	2067.6	12354
198	22.515	166.1	1145.0	34.97	16.582	47.64	66.99	2240.9	13409
199	22.987	167.1	1145.4	34.30	16.607	47.88	71.66	2440.9	14624
200	23.468	168.1	1145.8	33.64	16.632	48.12	77.14	2675.6	16052

Table 2. Thermodynamic Properties of Air, Ice and Steam

Temperature, F	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	v_s	h_a	h_s	W_s
32	0.1803	−143.4	1075.2	3305	12.389	7.69	12.46	11.75	26.40
31	0.1723	−143.9	1074.7	3453	12.363	7.45	12.43	11.32	25.21
30	0.1645	−144.4	1074.3	3608	12.338	7.21	12.41	10.90	24.07
29	0.1571	−144.9	1073.8	3771	12.313	6.97	12.38	10.49	22.98
28	0.1500	−145.4	1073.4	3943	12.287	6.73	12.35	10.09	21.92
27	0.1431	−145.9	1073.0	4122	12.262	6.49	12.32	9.70	20.92
26	0.1366	−146.4	1072.5	4311	12.237	6.25	12.29	9.31	19.96
25	0.1303	−146.9	1072.1	4509	12.211	6.01	12.27	8.92	19.04
24	0.1243	−147.4	1071.7	4717	12.186	5.77	12.24	8.55	18.16
23	0.1186	−147.9	1071.2	4936	12.161	5.53	12.21	8.18	17.32
22	0.1130	−148.4	1070.8	5166	12.136	5.29	12.18	7.81	16.51
21	0.1078	−148.9	1070.3	5408	12.110	5.05	12.15	7.45	15.73
20	0.1027	−149.4	1069.9	5662	12.085	4.81	12.13	7.10	14.99
19	$9.789\ (10)^{-2}$	−149.8	1069.5	5929	12.060	4.56	12.10	6.75	14.28
18	$9.326\ (10)^{-2}$	−150.3	1069.0	6210	12.035	4.32	12.07	6.40	13.61
17	$8.884\ (10)^{-2}$	−150.8	1068.6	6505	12.009	4.08	12.05	6.06	12.96
16	$8.461\ (10)^{-2}$	−151.3	1068.1	6817	11.984	3.84	12.02	5.73	12.34
15	$8.056\ (10)^{-2}$	−151.8	1067.7	7144	11.959	3.60	11.99	5.40	11.75
14	$7.669\ (10)^{-2}$	−152.3	1067.3	7489	11.933	3.36	11.96	5.07	11.18
13	$7.300\ (10)^{-2}$	−152.8	1066.8	7851	11.918	3.12	11.94	4.75	10.64
12	$6.946\ (10)^{-2}$	−153.3	1066.4	8234	11.883	2.88	11.91	4.43	10.13
11	$6.608\ (10)^{-2}$	−153.7	1065.9	8636	11.857	2.64	11.88	4.11	9.633
10	$6.286\ (10)^{-2}$	−154.2	1065.5	9060	11.832	2.40	11.86	3.80	9.161
9	$5.977\ (10)^{-2}$	−154.7	1065.1	9507	11.807	2.16	11.83	3.49	8.711
8	$5.683\ (10)^{-2}$	−155.2	1064.6	9979	11.782	1.92	11.80	3.18	8.280
7	$5.402\ (10)^{-2}$	−155.7	1064.2	$1.048\ (10)^{4}$	11.756	1.68	11.78	2.88	7.867
6	$5.134\ (10)^{-2}$	−156.1	1063.7	$1.100\ (10)^{4}$	11.731	1.44	11.75	2.58	7.478
5	$4.878\ (10)^{-2}$	−156.6	1063.3	$1.155\ (10)^{4}$	11.706	1.20	11.72	2.28	7.106
4	$4.633\ (10)^{-2}$	−157.1	1062.8	$1.214\ (10)^{4}$	11.680	0.96	11.70	1.98	6.745
3	$4.400\ (10)^{-2}$	−157.6	1062.4	$1.275\ (10)^{4}$	11.655	0.72	11.56	1.68	6.348
2	$4.178\ (10)^{-2}$	−158.0	1062.0	$1.340\ (10)^{4}$	11.630	0.48	11.65	1.40	6.084
1	$3.966\ (10)^{-2}$	−158.5	1061.5	$1.408\ (10)^{4}$	11.604	+ 0.24	11.62	1.12	5.777
0	$3.764\ (10)^{-2}$	−159.0	1061.1	$1.481\ (10)^{4}$	11.579	0.00	11.59	0.83	5.480
− 1	$3.572\ (10)^{-2}$	−159.4	1060.6	$1.577\ (10)^{4}$	11.554	− 0.24	11.57	0.55	5.201
− 2	$3.388\ (10)^{-2}$	−159.9	1060.2	$1.638\ (10)^{4}$	11.538	− 0.48	11.54	+ 0.27	4.932
− 3	$3.213\ (10)^{-2}$	−160.4	1059.8	$1.723\ (10)^{4}$	11.503	− 0.72	11.52	− 0.01	4.679
− 4	$3.047\ (10)^{-2}$	−160.9	1059.3	$1.814\ (10)^{4}$	11.478	− 0.96	11.49	− 0.29	4.434
− 5	$2.888\ (10)^{-2}$	−161.3	1058.9	$1.909\ (10)^{4}$	11.453	− 1.20	11.46	− 0.57	4.204
− 6	$2.737\ (10)^{-2}$	−161.8	1058.4	$2.010\ (10)^{4}$	11.427	− 1.44	11.44	− 0.84	3.983
− 7	$2.594\ (10)^{-2}$	−162.3	1058.0	$2.116\ (10)^{4}$	11.402	− 1.68	11.41	− 1.11	3.775
− 8	$2.457\ (10)^{-2}$	−162.7	1057.6	$2.229\ (10)^{4}$	11.377	− 1.92	11.39	− 1.38	3.576
− 9	$2.327\ (19)^{-2}$	−163.2	1057.1	$2.349\ (10)^{4}$	11.351	− 2.16	11.36	− 1.65	3.385
− 10	$2.203\ (10)^{-2}$	−163.6	1056.7	$2.475\ (10)^{4}$	11.326	− 2.40	11.33	− 1.92	3.206
− 11	$2.086\ (10)^{-2}$	−164.1	1056.2	$2.609\ (10)^{4}$	11.301	− 2.64	11.31	− 2.19	3.034
− 12	$1.974\ (10)^{-2}$	−164.6	1055.8	$2.750\ (10)^{4}$	11.276	− 2.88	11.28	− 2.45	2.872
− 13	$1.867\ (10)^{-2}$	−165.0	1055.4	$2.901\ (10)^{4}$	11.250	− 3.12	11.26	− 2.71	2.716
− 14	$1.766\ (10)^{-2}$	−165.5	1054.9	$3.060\ (10)^{4}$	11.225	− 3.37	11.23	− 2.98	2.569
− 15	$1.670\ (10)^{-2}$	−165.9	1054.5	$3.228\ (10)^{4}$	11.200	− 3.61	11.21	− 3.24	2.430
− 16	$1.579\ (10)^{-2}$	−166.4	1054.0	$3.407\ (10)^{4}$	11.174	− 3.85	11.18	− 3.50	2.297
− 17	$1.493\ (10)^{-2}$	−166.8	1053.6	$3.596\ (10)^{4}$	11.149	− 4.09	11.15	− 3.76	2.171
− 18	$1.410\ (10)^{-2}$	−167.3	1053.1	$3.797\ (10)^{4}$	11.123	− 4.33	11.13	− 4.02	2.052
− 19	$1.333\ (10)^{-2}$	−167.8	1052.7	$4.011\ (10)^{4}$	11.098	− 4.57	11.10	− 4.28	1.938
− 20	$1.259\ (10)^{-2}$	−168.2	1052.3	$4.237\ (10)^{4}$	11.073	− 4.81	11.08	− 4.53	1.830
− 21	$1.188\ (19)^{-2}$	−168.7	1051.8	$4.477\ (10)^{4}$	11.048	− 5.05	11.05	− 4.79	1.728
− 22	$1.122\ (10)^{-2}$	−169.1	1051.4	$4.732\ (10)^{4}$	11.022	− 5.29	11.03	− 5.04	1.631
− 23	$1.059\ (10)^{-2}$	−169.6	1050.9	$5.003\ (10)^{4}$	10.997	− 5.53	11.00	− 5.30	1.539
− 24	$9.987\ (10)^{-3}$	−170.0	1050.5	$5.290\ (10)^{4}$	10.972	− 5.77	10.98	− 5.55	1.452

Table 2. Thermodynamic Properties of Air, Ice and Steam (Continued)

	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
Temperature, F	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	h_a	v_s	h_s	W_s
−25	9.420 $(10)^{-3}$	−170.5	1050.1	5.596 $(10)^{4}$	10.947	−6.01	10.95	−5.80	1.370
−26	8.882 $(10)^{-3}$	−170.9	1049.6	5.921 $(10)^{4}$	10.921	−6.25	10.92	−6.05	1.292
−27	8.373 $(10)^{-3}$	−171.4	1049.2	6.267 $(10)^{4}$	10.896	−6.49	10.90	−6.31	1.217
−28	7.891 $(10)^{-3}$	−171.8	1048.7	6.634 $(10)^{4}$	10.871	−6.73	10.87	−6.56	1.147
−29	7.435 $(10)^{-3}$	−172.2	1048.3	7.025 $(10)^{4}$	10.845	−6.97	10.85	−6.81	1.081
−30	7.003 $(10)^{-3}$	−172.7	1047.8	7.441 $(10)^{4}$	10.820	−7.21	10.82	−7.06	1.018
−31	6.595 $(10)^{-3}$	−173.1	1047.4	7.883 $(10)^{4}$	10.795	−7.45	10.80	−7.31	0.9587
−32	6.208 $(10)^{-3}$	−173.6	1047.0	8.355 $(10)^{4}$	10.769	−7.69	10.77	−7.56	0.9024
−33	5.843 $(10)^{-3}$	−174.0	1046.5	8.857 $(10)^{4}$	10.744	−7.93	10.75	−7.80	0.8493
−34	5.497 $(10)^{-3}$	−174.4	1046.1	9.391 $(10)^{4}$	10.719	−8.17	10.72	−8.05	0.7991
−35	5.170 $(10)^{-3}$	−174.9	1045.6	9.961 $(10)^{4}$	10.693	−8.41	10.70	−8.30	0.7516
−36	4.862 $(10)^{-3}$	−175.3	1045.2	1.057 $(10)^{5}$	10.668	−8.65	10.67	−8.55	0.7066
−37	4.570 $(10)^{-3}$	−175.8	1044.7	1.122 $(10)^{5}$	10.642	−8.89	10.64	−8.79	0.6641
−38	4.295 $(10)^{-3}$	−176.2	1044.3	1.191 $(10)^{5}$	10.617	−9.13	10.62	−9.04	0.6241
−39	4.035 $(10)^{-3}$	−176.6	1043.9	1.264 $(10)^{5}$	10.592	−9.37	10.59	−9.29	0.5866
−40	3.790 $(10)^{-3}$	−177.1	1043.4	1.343 $(10)^{5}$	10.567	−9.61	10.57	−9.53	0.5508
−41	3.559 $(10)^{-3}$	−177.5	1043.0	1.427 $(10)^{5}$	10.541	−9.85	10.54	−9.78	0.5172
−42	3.340 $(10)^{-3}$	−177.9	1042.5	1.516 $(10)^{5}$	10.516	−10.10	10.52	−10.02	0.4856
−43	3.134 $(10)^{-3}$	−178.4	1042.1	1.612 $(10)^{5}$	10.491	−10.34	10.49	−10.27	0.4556
−44	2.940 $(10)^{-3}$	−178.8	1041.6	1.715 $(10)^{5}$	10.466	−10.58	10.47	−10.51	0.4272
−45	2.757 $(10)^{-3}$	−179.2	1041.2	1.824 $(10)^{5}$	10.440	−10.82	10.44	−10.76	0.4007
−46	2.585 $(10)^{-3}$	−179.6	1040.8	1.941 $(10)^{5}$	10.415	−11.06	10.42	−11.00	0.3756
−47	2.422 $(10)^{-3}$	−180.1	1040.3	2.066 $(10)^{5}$	10.389	−11.30	10.39	−11.24	0.3520
−48	2.270 $(10)^{-3}$	−180.5	1039.9	2.200 $(10)^{5}$	10.364	−11.54	10.37	−11.49	0.3298
−49	2.126 $(10)^{-3}$	−180.9	1039.4	2.343 $(10)^{5}$	10.339	−11.77	10.34	−11.73	0.3089
−50	1.990 $(10)^{-3}$	−181.3	1039.0	2.496 $(10)^{5}$	10.314	−12.01	10.31	−11.97	0.2893
−51	1.863 $(10)^{-3}$	−181.8	1038.6	2.660 $(10)^{5}$	10.288	−12.26	10.29	−12.22	0.2708
−52	1.743 $(10)^{-3}$	−182.2	1038.1	2.836 $(10)^{5}$	10.263	−12.50	10.26	−12.46	0.2533
−53	1.631 $(10)^{-3}$	−182.6	1037.7	3.024 $(10)^{5}$	10.238	−12.74	10.24	−12.70	0.2370
−54	1.525 $(10)^{-3}$	−183.0	1037.2	3.226 $(10)^{5}$	10.212	−12.98	10.21	−12.95	0.2216
−55	1.426 $(10)^{-3}$	−183.4	1036.8	3.443 $(10)^{5}$	10.187	−13.22	10.19	−13.19	0.2071
−56	1.332 $(10)^{-3}$	−183.9	1036.3	3.675 $(10)^{5}$	10.161	−13.46	10.16	−13.43	0.1936
−57	1.244 $(10)^{-3}$	−184.3	1035.9	3.925 $(10)^{5}$	10.136	−13.70	10.14	−13.67	0.1808
−58	1.162 $(10)^{-3}$	−184.7	1035.5	4.192 $(10)^{5}$	10.111	−13.94	10.11	−13.92	0.1688
−59	1.085 $(10)^{-3}$	−185.1	1035.0	4.479 $(10)^{5}$	10.086	−14.18	10.09	−14.16	0.1576
−60	1.012 $(10)^{-3}$	−185.5	1034.6	4.788 $(10)^{5}$	10.060	−14.42	10.06	−14.40	0.1471
−61	9.444 $(10)^{-4}$	−185.9	1034.1	5.120 $(10)^{5}$	10.035	−14.66	10.03	−14.64	0.1372
−62	8.808 $(10)^{-4}$	−186.4	1033.7	5.476 $(10)^{5}$	10.010	−14.90	10.01	−14.88	0.1280
−63	8.211 $(10)^{-4}$	−186.8	1033.2	5.859 $(10)^{5}$	9.984	−15.14	9.984	−15.12	0.1193
−64	7.652 $(10)^{-4}$	−187.2	1032.8	6.272 $(10)^{5}$	9.959	−15.38	9.959	−15.37	0.1112
−65	7.128 $(10)^{-4}$	−187.6	1032.4	6.715 $(10)^{5}$	9.933	−15.62	9.934	−15.61	0.1036
−66	6.638 $(10)^{-4}$	−188.0	1031.9	7.193 $(10)^{5}$	9.908	−15.86	9.908	−15.85	9.642 $(10)^{-2}$
−67	6.180 $(10)^{-4}$	−188.4	1031.5	7.707 $(10)^{5}$	9.883	−16.10	9.883	−16.09	8.976 $(10)^{-2}$
−68	5.751 $(10)^{-4}$	−188.8	1031.0	8.261 $(10)^{5}$	9.858	−16.34	9.858	−16.33	8.353 $(10)^{-2}$
−69	5.349 $(10)^{-4}$	−189.2	1030.6	8.858 $(10)^{5}$	9.832	−16.58	9.832	−16.57	7.770 $(10)^{-2}$
−70	4.974 $(10)^{-4}$	−189.6	1030.2	9.501 $(10)^{5}$	9.807	−16.82	9.807	−16.81	7.225 $(10)^{-2}$
−71	4.624 $(10)^{-4}$	−190.0	1029.7	1.020 $(10)^{6}$	9.781	−17.06	9.781	−17.06	6.713 $(10)^{-2}$
−72	4.296 $(10)^{-4}$	−190.4	1029.3	1.094 $(10)^{6}$	9.756	−17.31	9.756	−17.30	6.242 $(10)^{-2}$
−73	3.991 $(10)^{-4}$	−190.8	1028.8	1.175 $(10)^{6}$	9.731	−17.55	9.731	−17.54	5.797 $(10)^{-2}$
−74	3.705 $(10)^{-4}$	−191.2	1028.4	1.262 $(10)^{6}$	9.705	−17.79	9.705	−17.78	5.383 $(10)^{-2}$
−75	3.439 $(10)^{-4}$	−191.6	1027.9	1.356 $(10)^{6}$	9.680	−18.03	9.680	−18.02	4.997 $(10)^{-2}$
−76	3.191 $(10)^{-4}$	−192.0	1027.5	1.458 $(10)^{6}$	9.655	−18.27	9.655	−18.26	4.635 $(10)^{-2}$
−77	2.959 $(10)^{-4}$	−192.4	1027.1	1.568 $(10)^{6}$	9.629	−18.51	9.629	−18.50	4.299 $(10)^{-2}$
−78	2.743 $(10)^{-4}$	−192.8	1026.6	1.687 $(10)^{6}$	9.604	−18.75	9.604	−18.74	3.985 $(10)^{-2}$
−79	2.542 $(10)^{-4}$	−193.2	1026.2	1.816 $(10)^{6}$	9.579	−18.99	9.579	−18.99	3.692 $(10)^{-2}$
−80	2.355 $(10)^{-4}$	−193.6	1025.7	1.955 $(10)^{6}$	9.553	−19.23	9.553	−19.23	3.421 $(10)^{-2}$
−81	2.181 $(10)^{-4}$	−194.0	1025.3	2.106 $(10)^{6}$	9.528	−19.47	9.528	−19.47	3.167 $(10)^{-2}$
−82	2.018 $(10)^{-4}$	−194.4	1024.8	2.270 $(10)^{6}$	9.503	−19.71	9.502	−19.71	2.930 $(10)^{-2}$
−83	1.867 $(10)^{-4}$	−194.8	1024.4	2.446 $(10)^{6}$	9.477	−19.95	9.477	−19.95	2.712 $(10)^{-2}$
−84	1.727 $(10)^{-4}$	−195.2	1024.0	2.638 $(10)^{6}$	9.472	−20.19	9.452	−20.19	2.508 $(10)^{-2}$

Table 2. Thermodynamic Properties of Air, Ice and Steam (Continued)

	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in.Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
Temperature, F	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	h_a	v_s	h_s	W_s
− 85	$1.596\,(10)^{-4}$	−195.6	1023.5	$2.846\,(10)^{6}$	9.426	−20.43	9.426	−20.43	$2.318\,(10)^{-2}$
− 86	$1.475\,(10)^{-4}$	−195.9	1023.1	$3.072\,(10)^{6}$	9.401	−20.67	9.401	−20.67	$2.142\,(10)^{-2}$
− 87	$1.363\,(10)^{-4}$	−196.3	1022.6	$3.317\,(10)^{6}$	9.376	−20.91	9.376	−20.91	$1.979\,(10)^{-2}$
− 88	$1.258\,(10)^{-4}$	−196.7	1022.2	$3.583\,(10)^{6}$	9.350	−21.15	9.350	−21.15	$1.827\,(10)^{-2}$
− 89	$1.161\,(10)^{-4}$	−197.1	1021.8	$3.872\,(10)^{6}$	9.325	−21.40	9.325	−21.40	$1.686\,(10)^{-2}$
− 90	$1.071\,(10)^{-4}$	−197.5	1021.3	$4.186\,(10)^{6}$	9.299	−21.64	9.300	−21.64	$1.555\,(10)^{-2}$
− 91	$9.876\,(10)^{-5}$	−197.9	1020.9	$4.528\,(10)^{6}$	9.274	−21.88	9.274	−21.88	$1.434\,(10)^{-2}$
− 92	$9.102\,(10)^{-5}$	−198.2	1020.4	$4.899\,(10)^{6}$	9.249	−22.12	9.249	−22.12	$1.322\,(10)^{-2}$
− 93	$8.385\,(10)^{-5}$	−198.6	1020.0	$5.303\,(10)^{6}$	9.224	−22.36	9.224	−22.36	$1.218\,(10)^{-2}$
− 94	$7.722\,(10)^{-5}$	−199.0	1019.5	$5.743\,(10)^{6}$	9.198	−22.60	9.198	−22.60	$1.121\,(10)^{-2}$
− 95	$7.108\,(10)^{-5}$	−199.4	1019.1	$6.223\,(10)^{6}$	9.173	−22.84	9.173	−22.84	$1.032\,(10)^{-2}$
− 96	$6.539\,(10)^{-5}$	−199.8	1018.7	$6.745\,(10)^{6}$	9.147	−23.08	9.147	−23.08	$9.493\,(10)^{-3}$
− 97	$6.014\,(10)^{-5}$	−200.2	1018.2	$7.314\,(10)^{6}$	9.122	−23.32	9.122	−23.32	$8.730\,(10)^{-3}$
− 98	$5.528\,(10)^{-5}$	−200.5	1017.8	$7.936\,(10)^{6}$	9.097	−23.56	9.097	−23.56	$8.023\,(10)^{-3}$
− 99	$5.079\,(10)^{-5}$	−200.9	1017.3	$8.613\,(10)^{6}$	9.071	−23.80	9.071	−23.80	$7.372\,(10)^{-3}$
−100	$4.664\,(10)^{-5}$	−201.3	1016.9	$9.352\,(10)^{6}$	9.046	−24.04	9.046	−24.04	$6.771\,(10)^{-3}$
−101	$4.281\,(10)^{-5}$	−201.6	1016.5	$1.016\,(10)^{7}$	9.020	−24.28	9.020	−24.28	$6.215\,(10)^{-3}$
−102	$3.928\,(10)^{-5}$	−202.0	1016.1	$1.104\,(10)^{7}$	8.995	−24.52	8.995	−24.52	$5.703\,(10)^{-3}$
−103	$3.602\,(10)^{-5}$	−202.4	1015.6	$1.201\,(10)^{7}$	8.970	−24.77	8.970	−24.77	$5.228\,(10)^{-3}$
−104	$3.301\,(10)^{-5}$	−202.8	1015.1	$1.307\,(10)^{7}$	8.944	−25.01	8.944	−25.01	$4.790\,(10)^{-3}$
−105	$3.025\,(10)^{-5}$	−203.1	1014.7	$1.422\,(10)^{7}$	8.919	−25.25	8.919	−25.25	$4.390\,(10)^{-3}$
−106	$2.770\,(10)^{-5}$	−203.5	1014.2	$1.549\,(10)^{7}$	8.894	−25.49	8.894	−25.49	$4.019\,(10)^{-3}$
−107	$2.535\,(10)^{-5}$	−203.9	1013.8	$1.687\,(10)^{7}$	8.868	−25.73	8.868	−25.73	$3.680\,(10)^{-3}$
−108	$2.319\,(10)^{-5}$	−204.2	1013.4	$1.839\,(10)^{7}$	8.843	−25.97	8.843	−25.97	$3.366\,(10)^{-3}$
−109	$2.120\,(10)^{-5}$	−204.6	1012.9	$2.006\,(10)^{7}$	8.817	−26.21	8.817	−26.21	$3.077\,(10)^{-3}$
−110	$1.938\,(10)^{-5}$	−205.0	1012.5	$2.189\,(10)^{7}$	8.792	−26.45	8.792	−26.45	$2.812\,(10)^{-3}$
−111	$1.770\,(10)^{-5}$	−205.3	1012.0	$2.389\,(10)^{7}$	8.767	−26.69	8.766	−26.69	$2.569\,(10)^{-3}$
−112	$1.616\,(10)^{-5}$	−205.7	1011.6	$2.610\,(10)^{7}$	8.741	−26.93	8.741	−26.93	$2.344\,(10)^{-3}$
−113	$1.476\,(10)^{-5}$	−206.1	1011.1	$2.852\,(10)^{7}$	8.716	−27.17	8.716	−27.17	$2.139\,(10)^{-3}$
−114	$1.344\,(10)^{-5}$	−206.4	1010.7	$3.118\,(10)^{7}$	8.690	−27.41	8.690	−27.41	$1.951\,(10)^{-3}$
−115	$1.226\,(10)^{-5}$	−206.8	1010.3	$3.411\,(10)^{7}$	8.665	−27.66	8.665	−27.66	$1.778\,(10)^{-3}$
−116	$1.116\,(10)^{-5}$	−207.2	1009.8	$3.733\,(10)^{7}$	8.640	−27.90	8.640	−27.90	$1.620\,(10)^{-3}$
−117	$1.017\,(10)^{-5}$	−207.5	1009.4	$4.088\,(10)^{7}$	8.614	−28.14	8.614	−28.14	$1.475\,(10)^{-3}$
−118	$9.251\,(10)^{-6}$	−207.9	1008.9	$4.480\,(10)^{7}$	8.589	−28.38	8.589	−28.38	$1.342\,(10)^{-3}$
−119	$8.414\,(10)^{-6}$	−208.2	1008.5	$4.911\,(10)^{7}$	8.563	−28.62	8.563	−28.62	$1.221\,(10)^{-3}$
−120	$7.649\,(10)^{-6}$	−208.6	1008.0	$5.386\,(10)^{7}$	8.538	−28.86	8.538	−28.86	$1.110\,(10)^{-3}$
−121	$6.949\,(10)^{-6}$	−208.9	1007.6	$5.911\,(10)^{7}$	8.512	−29.10	8.512	−29.10	$1.008\,(10)^{-3}$
−122	$6.310\,(10)^{-6}$	−209.3	1007.2	$6.491\,(10)^{7}$	8.487	−29.34	8.487	−29.34	$9.152\,(10)^{-4}$
−123	$5.726\,(10)^{-6}$	−209.6	1006.7	$7.131\,(10)^{7}$	8.462	−29.58	8.462	−29.58	$8.306\,(10)^{-4}$
−124	$5.194\,(10)^{-6}$	−210.0	1006.3	$7.839\,(10)^{7}$	8.436	−29.82	8.436	−29.82	$7.533\,(10)^{-4}$
−125	$4.708\,(10)^{-6}$	−210.3	1005.8	$8.622\,(10)^{7}$	8.411	−30.06	8.411	−30.06	$6.829\,(10)^{-4}$
−126	$4.265\,(10)^{-6}$	−210.7	1005.4	$9.489\,(10)^{7}$	8.386	−30.30	8.386	−30.30	$6.186\,(10)^{-4}$
−127	$3.862\,(10)^{-6}$	−211.0	1005.0	$1.045\,(10)^{8}$	8.360	−30.55	8.360	−30.55	$5.600\,(10)^{-4}$
−128	$3.494\,(10)^{-6}$	−211.4	1004.5	$1.151\,(10)^{8}$	8.335	−30.79	8.335	−30.79	$5.069\,(10)^{-4}$
−129	$3.160\,(10)^{-6}$	−211.7	1004.1	$1.269\,(10)^{8}$	8.309	−31.03	8.309	−31.03	$4.584\,(10)^{-4}$
−130	$2.856\,(10)^{-6}$	−212.1	1003.6	$1.400\,(10)^{2}$	8.284	−31.27	8.284	−31.27	$4.142\,(10)^{-4}$
−131	$2.579\,(10)^{-6}$	−212.4	1003.2	$1.545\,(10)^{8}$	8.258	−31.51	8.258	−31.51	$3.742\,(10)^{-4}$
−132	$2.328\,(10)^{-6}$	−212.8	1002.8	$1.707\,(10)^{8}$	8.233	−31.75	8.233	−31.75	$3.376\,(10)^{-4}$
−133	$2.101\,(10)^{-6}$	−213.1	1002.3	$1.886\,(10)^{8}$	8.207	−31.99	8.207	−31.99	$3.046\,(10)^{-4}$
−134	$1.894\,(10)^{-6}$	−213.5	1001.9	$2.086\,(10)^{8}$	8.182	−32.23	8.182	−32.23	$2.746\,(10)^{-4}$
−135	$1.706\,(10)^{-6}$	−213.8	1001.4	$2.308\,(10)^{8}$	8.157	−32.47	8.157	−32.47	$2.474\,(10)^{-4}$
−136	$1.536\,(10)^{-6}$	−214.1	1001.0	$2.555\,(10)^{8}$	8.131	−32.71	8.131	−32.71	$2.228\,(10)^{-4}$
−137	$1.382\,(10)^{-6}$	−214.5	1000.5	$2.831\,(10)^{8}$	8.106	−32.96	8.106	−32.96	$2.004\,(10)^{-4}$
−138	$1.243\,(10)^{-6}$	−214.8	1000.1	$3.138\,(10)^{8}$	8.080	−33.20	8.080	−33.20	$1.802\,(10)^{-4}$
−139	$1.117\,(10)^{-6}$	−215.2	999.7	$3.481\,(10)^{8}$	8.055	−33.44	8.055	−33.44	$1.620\,(10)^{-4}$
−140	$1.003\,(10)^{-6}$	−215.5	999.2	$3.864\,(10)^{8}$	8.029	−33.68	8.029	−33.68	$1.455\,(10)^{-4}$
−141	$9.005\,(10)^{-7}$	−215.8	998.8	$4.292\,(10)^{8}$	8.004	−33.92	8.004	−33.92	$1.305\,(10)^{-4}$
−142	$8.076\,(10)^{-7}$	−216.2	998.3	$4.770\,(10)^{8}$	7.978	−34.16	7.978	−34.16	$1.171\,(10)^{-4}$
−143	$7.240\,(10)^{-7}$	−216.5	997.9	$5.305\,(10)^{8}$	7.953	−34.40	7.953	−34.40	$1.049\,(10)^{-4}$
−144	$6.483\,(10)^{-7}$	−216.8	997.4	$5.905\,(10)^{8}$	7.928	−34.65	7.928	−34.65	$9.397\,(10)^{-5}$

Table 2. Thermodynamic Properties of Air, Ice and Steam (Concluded)

Temperature, F	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in.Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	h_a	v_s	h_s	W'_s
−145	$5.803\ (10)^{-7}$	−217.2	997.0	$6.577\ (10)^{8}$	7.902	−34.89	7.902	−34.89	$8.410\ (10)^{-5}$
−146	$5.190\ (10)^{-7}$	−217.5	996.6	$7.330\ (10)^{8}$	7.876	−35.13	7.877	−35.13	$7.522\ (10)^{-5}$
−147	$4.639\ (100^{-7}$	−217.8	996.1	$8.174\ (10)^{8}$	7.851	−35.37	7.851	−35.37	$6.724\ (10)^{-5}$
−148	$4.143\ (10)^{-7}$	−218.2	995.7	$9.123\ (10)^{8}$	7.826	−35.61	7.826	−35.61	$6.005\ (10)^{-5}$
−149	$3.698\ (10)^{-7}$	−218.5	995.2	$10.19\ (10)^{8}$	7.800	−35.85	7.800	−35.85	$5.358\ (10)^{-5}$
−150	$3.298\ (10)^{-7}$	−218.8	994.8	$11.39\ (10)^{8}$	7.775	−36.09	7.775	−36.09	$4.778\ (10)^{-5}$
−151	$2.939\ (10)^{-7}$	−219.1	994.4	$12.74\ (10)^{8}$	7.749	−36.33	7.749	−36.33	$4.258\ (10)^{-5}$
−152	$2.618\ (10)^{-7}$	−219.5	993.9	$14.26\ (10)^{8}$	7.724	−36.58	7.724	−36.58	$3.792\ (10)^{-5}$
−153	$2.329\ (10)^{-7}$	−219.8	993.5	$15.97\ (10)^{8}$	7.698	−36.82	7.698	−36.82	$3.374\ (10)^{-5}$
−154	$2.072\ (10)^{-7}$	−220.1	993.0	$17.90\ (10)^{8}$	7.673	−37.06	7.673	−37.06	$3.001\ (10)^{-5}$
−155	$1.840\ (10)^{-7}$	−220.4	992.6	$20.08\ (10)^{8}$	7.647	−37.30	7.647	−37.30	$2.666\ (10)^{-5}$
−156	$1.634\ (10)^{-7}$	−220.8	992.1	$22.54\ (10)^{8}$	7.622	−37.54	7.622	−37.54	$2.367\ (10)^{-5}$
−157	$1.450\ (10)^{-7}$	−221.1	991.7	$25.32\ (10)^{8}$	7.596	−37.78	7.597	−37.78	$2.100\ (10)^{-5}$
−158	$1.285\ (10)^{-7}$	−221.4	991.3	$28.47\ (10)^{8}$	7.571	−38.02	7.571	−38.02	$1.861\ (10)^{-5}$
−159	$1.138\ (10)^{-7}$	−221.7	990.8	$32.03\ (10)^{8}$	7.546	−38.27	7.546	−38.27	$1.649\ (10)^{-5}$
−160	$1.008\ (10)^{-7}$	−222.0	990.4	$36.07\ (10)^{8}$	7.520	−38.51	7.520	−38.51	$1.459\ (10)^{-5}$

Table 3. Condensed Table of Thermodynamic Properties of Air, Water (Ice) and Steam (Giving Magnitudes of Pressure Corrections)*

Temperature, F	Properties of Dry Air at a Pressure of 29.921 in. Hg abs						Properties of Water (Ice) and Steam				Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Sp vol, cu ft/lb (varies with pressure)	Sp vol, cu ft/lb correction (independent of pressure)	True specific volume, cu ft/lb	Enthalpy, Btu/lb at zero pressure	Enthalpy, Btu/lb correction (varies with pressure)	Enthalpy, Btu/lb	Saturation pressure of ice (water) and steam, in. Hg	Enthalpy of sat ice (water), Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	Volume of mixture per lb of dry air, cu ft/lb	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
	v_1	v_2	v_a	h_1	h_1	h_a	p_s	h_i (or) h_f	h_g	v_g	v_s	h_s	W_s
Part I. Conditions over Ice													
−150	7.808	0.033	7.775	−35.78	0.31	−36.09	$3.298(10)^{-7}$	−218.8	994.8	$11.39(10)^{8}$	7.775	−36.09	$4.778(10)^{-5}$
−140	8.060	0.031	8.029	−33.39	0.29	−33.68	$1.003(10)^{-6}$	−215.5	999.2	$3.864(10)^{8}$	8.029	−33.68	$1.455(10)^{-4}$
−130	8.312	0.028	8.284	−31.00	0.27	−31.27	$2.856(10)^{-6}$	−212.1	1003.6	$1.400(10)^{8}$	8.284	−31.27	$4.142(10)^{-4}$
−120	8.564	0.026	8.538	−28.60	0.26	−28.86	$7.649(10)^{-6}$	−208.6	1008.0	$5.386(10)^{7}$	8.538	−28.86	$1.110(10)^{-3}$
−110	8.816	0.024	8.792	−26.21	0.24	−26.45	$1.938(10)^{-5}$	−205.0	1012.5	$2.189(10)^{7}$	8.792	−26.45	$2.812(10)^{-3}$
−100	9.068	0.022	9.046	−23.81	0.23	−24.04	$4.664(10)^{-5}$	−201.3	1016.9	$9.352(10)^{6}$	9.046	−24.04	$6.771(10)^{-3}$
− 90	9.320	0.021	9.299	−21.42	0.22	−21.64	$1.071(10)^{-4}$	−197.5	1021.3	$4.186(10)^{6}$	9.300	−21.64	$1.555(10)^{-2}$
− 80	9.572	0.019	9.553	−19.03	0.20	−19.23	$2.355(10)^{-4}$	−193.6	1025.7	$1.955(10)^{6}$	9.553	−19.23	$3.421(10)^{-2}$
− 70	9.824	0.017	9.807	−16.63	0.19	−16.82	$4.974(10)^{-4}$	−189.6	1030.2	$9.501(10)^{5}$	9.807	−16.81	$7.225(10)^{-2}$
− 60	10.076	0.016	10.060	−14.24	0.18	−14.42	$1.012(10)^{-3}$	−185.5	1034.6	$4.788(10)^{5}$	10.06	−14.40	0.1471
− 50	10.329	0.015	10.314	−11.84	0.17	−12.01	$1.990(10)^{-3}$	−181.3	1039.0	$2.496(10)^{5}$	10.31	−11.97	0.2893
− 40	10.581	0.014	10.567	− 9.45	0.16	− 9.61	$3.790(10)^{-3}$	−177.1	1043.4	$1.343(10)^{5}$	10.57	− 9.53	0.5508
− 30	10.833	0.013	10.820	− 7.05	0.16	− 7.21	$7.003(10)^{-3}$	−172.7	1047.8	$7.441(10)^{4}$	10.82	− 7.06	1.018
− 20	11.085	0.012	11.073	− 4.66	0.15	− 4.81	$1.259(10)^{-2}$	−168.2	1052.3	$4.237(10)^{4}$	11.08	− 4.53	1.830
− 10	11.337	0.011	11.326	− 2.26	0.14	− 2.40	$2.203(10)^{-2}$	−163.6	1056.7	$2.475(10)^{4}$	11.33	− 1.92	3.206
0	11.589	0.010	11.579	+ 0.14	0.14	0.00	$3.764(10)^{-2}$	−159.0	1061.1	$1.481(10)^{4}$	11.59	0.83	5.480
10	11.841	0.009	11.832	2.53	0.13	2.40	$6.286(10)^{-2}$	−154.2	1065.5	9060	11.86	3.80	9.161
20	12.093	0.008	12.085	4.93	0.12	4.81	0.1027	−149.4	1069.9	5662	12.13	7.10	14.99
30	12.345	0.007	12.338	7.33	0.12	7.21	0.1645	−144.4	1074.3	3608	12.41	10.90	24.07
32	12.396	0.007	12.389	7.81	0.12	7.69	0.1803	−143.4	1075.2	3305	12.46	11.75	26.40
Part II. Conditions over Water													
32	12.396	0.007	12.389	7.81	0.12	7.69	0.1803	0.0	1075.2	3305	12.46	11.75	26.40
40	12.598	0.007	12.591	9.73	0.12	9.61	0.2477	8.0	1078.7	2445	12.70	15.21	36.34
50	12.850	0.006	12.844	12.13	0.11	12.02	0.3624	18.1	1083.1	1704	13.00	20.28	53.40
60	13.101	0.005	13.096	14.53	0.11	14.42	0.5216	28.1	1087.4	1207	13.33	26.43	77.29
70	13.354	0.005	13.349	16.93	0.10	16.83	0.7392	38.1	1091.8	868.0	13.69	34.04	110.4
80	13.606	0.004	13.602	19.33	0.10	19.23	1.0323	48.1	1096.1	633.0	14.09	43.63	155.8
90	13.858	0.004	13.854	21.73	0.09	21.64	1.4219	58.0	1100.4	467.9	14.55	55.85	217.6
100	14.110	0.003	14.107	24.13	0.09	24.04	1.9334	68.0	1104.7	350.2	15.08	71.62	301.5
110	14.362	0.002	14.360	26.54	0.09	26.45	2.5968	78.0	1109.0	265.3	15.72	92.19	414.9
120	14.614	0.002	14.612	28.93	0.08	28.85	3.4477	88.0	1113.3	203.2	16.51	119.36	569.0
130	14.866	0.001	14.865	31.34	0.08	31.26	4.5274	98.0	1117.5	157.3	17.52	155.72	779.6
140	15.118	0.001	15.117	33.75	0.08	33.67	5.8842	108.0	1121.7	123.0	18.82	205.34	1071
150	15.371	0.001	15.370	36.14	0.07	36.07	7.5727	118.0	1125.8	97.04	20.58	274.84	1485
160	15.622	0.000	15.622	38.55	0.07	38.48	9.6556	128.0	1129.9	77.27	23.07	375.81	2090
170	15.875	0.000	15.875	40.96	0.07	40.89	12.203	138.0	1133.9	62.04	26.81	530.86	3024
180	16.127	−0.001	16.128	43.37	0.07	43.30	15.295	148.0	1137.9	50.22	32.99	790.88	4598
190	16.379	−0.001	16.380	45.77	0.06	45.71	19.017	158.0	1141.9	40.96	44.94	1298.9	7682
200	16.631	−0.001	16.632	48.18	0.06	48.12	23.468	168.1	1145.8	33.64	77.14	2675.6	16052

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* For more complete values of moist air, see Tables 1 and 2.

11. WEATHER AND WATER DATA

THE data included in this chapter are limited in their scope to the information necessary to permit general engineering work and the design of both summer and winter air conditioning systems. Water temperature data are applicable for use in general industrial design as well as in refrigeration equipment selection.

The tabular data in this chapter are divided into three main classifications—winter weather conditions, summer weather conditions, and water temperature data for summer.

Winter Weather Conditions

Winter weather conditions are based on certain temperatures occurring during the colder periods of the normal heating season. There is a wide variation in the duration of the heating season among various localities or sections of the country. The variation is from approximately 135 days in certain parts of Arizona, Florida, and Texas to 365 days, or all year round, in San Francisco.

The duration of the heating season, however, is not necessarily a criterion of heating requirements, since San Francisco with a year-round heating season has a mean temperature of 56 F and a resultant average degree-day figure of 2,500 annually, while Memphis, Tenn., with a heating season of about 195 days, has a mean temperature of about 50 F during this period, with a corresponding degree-day figure of more than 3,000.

The choice of a design temperature in any locality is a matter of judgment and experience. Several methods for establishing winter outside design temperatures have been proposed and used successfully, although there is not exact agreement among these methods.

The most important factors influencing the design of heating or winter air conditioning are outside design temperature and wind direction and velocity. The choice of the design temperature is probably most logically based on the frequency of its occurrence. Other interesting data are elevation, latitude, lowest temperature on record, and average annual minimum temperature. The latter temperature is obtained by averaging the lowest temperatures (one for each year) recorded for the period of years the local weather bureau has been in operation.

Table 1 lists the data outlined in the foregoing paragraph for representative cities in each state of the United States and some data on Canadian cities.

Elevations given in column A, Table 1, are based on information from the "Dictionary of Elevations" issued by the United States Geodetic Survey. The elevations shown are not necessarily at the weather stations for the same localities. In many instances, the stations are located some distance from the city and at higher elevations. Wherever possible, the elevations are those listed at public buildings in the city proper, such as municipal buildings or court houses. In a few cases, railroad elevations at local depots have been tabulated.

Latitude in degrees and minutes is given in column B, Table 1. Most of these data are from previously published sources. The Canadian latitudes have been approximated from atlas maps.

Column C, Table 1, shows the lowest temperature on record from the United States Department of Agriculture's "Yearbook of Agriculture," 1941 Edition.

The average annual minimum temperatures shown in column D, Table 1, are taken from official U. S. Weather Bureau records, in most cases representing an average of all data available since the establishment of the weather station.

The recommended design winter outside dry-bulb temperature is given in column E.

Table 1. Winter Weather Conditions

(All temperatures in deg F)

State	City	A	B	C	D	E	F	G
		Elevation	Latitude degrees and minutes	Lowest temp on record	Average annual min temp	Recommended design temp	Average winter wind direction and velocity	Design temperature frequency occurrence*
Ala.								
	Birmingham	694	33-32	−10	12	10	N— 8.0	20- 5
	Mobile	15	30-42	− 1	22	20	N— 10.2	17- 3
Ariz.								
	Flagstaff	6,907	35-12	−30	−15	−15	SW— 7.8	20- 5
	Phoenix	1,083	33-28	16	26	25	E— 5.3	15- 3
Ark.								
	Fort Smith	448	35-32	−15	6	5	E— 8.1	23- 7
	Little Rock	288	34-45	−12	10	5	NW— 8.3	8- 4
Calif.								
	Los Angeles	286	34- 3	28	37	35	NE— 6.3	15- 6
	San Francisco	6	37-47	27	37	35	N— 7.5	27- 7
Colo.								
	Denver	5,183	39 45	−29	−11	−10	S— 7.5	33- 9
	Grand Junction	4,573	39- 4	−21	2	−10	NW— 4.3	40- 9
Conn.								
	New Haven	11	41 18	−15	− 1	− 5	N— 9.5	14- 5
Del.								
	Dover	34	39 8	−11	10	5	NW 14	—
	Wilmington	8	39 43	−15	6	0	NW—12	—
D. C.								
	Washington	5	38 54	−15	− 1	0	NW— 7.9	—
Fla.								
	Jacksonville	8	30 20	10	29	25	NE— 9.1	24- 5
	Tampa	15	27 57	19	30	35	NE— 8	34- 6
Ga.								
	Atlanta	1,050	33-39	− 8	12	10	NW—11.4	20- 3
	Savannah	21	32 5	8	22	15	NW— 9.5	3- 2
Idaho								
	Lewiston	756	46-25	−23	1	− 5	E 5.3	48- 9
	Pocatello	4,468	42-52	−22	−12	−10	SE 9	52- 9
Ill.								
	Chicago	600	41-47	−23	− 8	−10	W 12.5	18- 6
	Springfield	612	39-48	−24	− 7	−10	NW—12.1	15- 3
Ind.								
	Evansville	383	37-58	−16	1	0	S - 9.8	34-10
	Fort Wayne	788	41- 5	−24	−10	− 5	SW—10.5	54-13
	Indianapolis	708	39-46	−25	− 6	− 5	S— 11.5	42-12
Iowa								
	Dubuque	647	42-30	−32	−17	−15	NW— 7.1	46- 9
	Sioux City	1,107	42-30	−35	−20	−20	NW—11.6	23- 7

Table 1. Winter Weather Conditions (Continued)

State	City	A	B	C	D	E	F	G
		Elevation	Latitude degrees and minutes	Lowest temp on record	Average annual min temp	Recommended design temp	Average winter wind direction and velocity	Design temperature frequency occurrence*
Kan.	Concordia	1,375	39 35	−25	−13	−10	N— 7.7	24-6
	Dodge City	2,522	37 45	−26	−10	−10	NW 10.3	16-4
Ky.	Louisville	449	48-15	−20	− 5	0	SW 9.9	—
La.	New Orleans	2	29 57	7	26	20	NE 8.7	4-2
	Shreveport	204	32-30	− 5	16	10	SE 8.1	5-2
Maine	Eastport	33	44-54	−23	−15	−10	NW—12.6	40-8
	Portland	14	43-39	−21	− 6	−10	NW— 9.2	18-5
Md.	Baltimore	20	39 17	− 7	8	0	SW— 8.1	4-2
Mass.	Boston	10	42 21	−18	− 3	− 5	W— 11.2	16-5
Mich.	Alpena	587	45 4	−26	−12	−15	W— 11.0	16-5
	Grand Rapids	638	42-58	−24	−10	−10	W— 9.0	9-3
	Detroit	583	42-24	−24	−11	−10	SW—12.7	8-3
	Marquette	652	46-34	−27	−13	−15	NW—10.8	21-5
Minn.	Duluth	609	46 47	−41	−28	−30	SW—13.5	21-5
	Minneapolis	812	44 59	−34	−23	−20	NW—11.3	46-11
Miss	Vicksburg	100	32 22	− 1	18	15	SE— 8.3	17-6
Mo.	St. Joseph	957	39 49	−24	−12	−15	W— 9.3	11-3
	St. Louis	460	38-38	−22	− 2	− 5	S— 11.7	16-6
	Springfield	1,301	37-12	−29	− 5	− 5	SE— 10.9	23-6
Mont.	Billings	3,117	45-43	−49	−30	−30	SW—11.9	27-4
	Havre	2,480	48-34	−57	−36	−40	SW— 9.5	9-5
Neb.	Lincoln	1,180	40-49	−29	−13	−15	S— 10.7	28-9
	North Platte	2,805	41- 8	−35	−17	−20	W— 8.3	14-3
Nev.	Reno	4,490	39-32	−19	−10	− 5	W— 6.1	17-6
	Tonopah	6,033	38- 6	−15	− 2	0	SE— 10.0	18-5
	Winnemucca	4,287	40-58	−28	−10	−15	NE— 8.2	36-9
N. Hamp.	Concord	244	43-12	−32	−15	−15	NW— 6.6	28-7
N. J.	Atlantic City	10	39-22	− 9	6	0	NW—15.6	6-5

Table 1. Winter Weather Conditions (Continued)

		A	B	C	D	E	F	G
State	City	Elevation	Latitude degrees and minutes	Lowest temp on record	Average annual min temp	Recommended design temp	Average winter wind direction and velocity	Design temperature frequency occurrence*
N. Y.	Albany	16	42–39	−24	−11	−10	S— 8.4	34– 9
	Buffalo	580	42–53	−20	− 4	− 5	W— 17.2	21– 7
	New York	10	40–43	−14	− 3	− 5	NW—16.7	8– 5
N. Mex.	Santa Fe	6,950	35–41	−13	0	0	NE— 7.8	27– 4
N. C.	Raleigh	316	35–45	− 2	13	15	SW— 7.9	46–11
	Wilmington	6	34–14	5	18	20	W— 8.1	54– 8
N. D.	Bismarck	1,672	46–48	−45	−31	−30	NW— 9.0	38– 6
	Devils Lake	1,471	48– 7	−46	−33	−30	W— 10.4	67–14
Ohio	Cincinnati	490	39– 9	−17	− 5	− 5	SW— 8.3	27– 9
	Cleveland	582	41–30	−17	− 2	− 5	SW—15	19– 5
	Columbus	780	39–58	−20	− 3	− 5	SW—11.6	30–10
Okla.	Okla. City	1,197	35–26	−17	2	0	NW—11.8	19– 6
Ore.	Baker	3,435	44–46	−24	−17	−15	SE— 6.8	23– 5
	Portland	29	45–32	− 2	18	10	S— 7.4	3– 3
Penna.	Philadelphia	9	39–57	−11	6	0	NW—11	6– 3
	Pittsburgh	744	40–21	−20	− 2	− 5	W— 11.7	14– 7
	Scranton	737	41–24	−19	−10	− 5	SW— 7.9	20– 7
R. I.	Providence	11	41–50	−17	1	− 5	NW—12.6	13– 5
S. C.	Charleston	10	32–47	7	22	20	SW—10.6	16– 6
	Columbia	312	34– 0	− 2	19	15	NE— 8.1	16– 5
S. D.	Huron	1,282	44–21	−43	−26	−20	NW—10.5	62–14
	Rapid City	3,231	44– 4	−40	−21	−20	W— 7.9	52–11
Tenn.	Knoxville	890	35–58	−16	2	5	SW— 7.1	18– 4
	Memphis	270	35– 9	− 9	9	5	S— 9.4	10– 4
Tex.	El Paso	3,695	31–47	− 5	16	15	NW— 9.0	8– 3
	Ft. Worth	616	32–45	− 8	12	10	NW—10.3	12– 3
	San Antonio	654	29–27	4	21	20	NE— 8.3	18– 5
Utah	Modena	5,460	37–48	−27	−15	−15	W— 9.3	37– 9
	Salt Lake City	4,220	40–46	−20	2	− 5	SE— 6.7	15– 6
Vt.	Burlington	112	44–29	−30	−17	−15	S— 11.7	60–12

Table 1. Winter Weather Conditions (Concluded)

State	City	A	B	C	D	E	F	G
		Elevation	Latitude degrees and minutes	Lowest temp on record	Average annual min temp	Recommended design temp	Average winter wind direction and velocity	Design temperature frequency occurrence*
Va.	Lynchburg	686	37–25	− 7	8	5	NW— 8.1	25– 7
	Norfolk	12	36–51	2	15	15	N— 12.4	34– 8
	Richmond	20	37–32	− 3	10	10	SW— 8.0	—
Wash.	Seattle	10	47–36	3	20	15	SE— 10.1	11– 3
	Spokane	1,890	47–40	−30	− 5	−15	SW— 6.3	7– 2
W. Va.	Elkins	1,940	38–53	−28	− 8	−10	W— 5.8	15– 3
	Parkersburg	615	39–16	−27	− 1	− 5	SW— 7.3	17– 7
Wis.	Green Bay	589	44–31	−36	−18	−20	SW—10.6	21– 4
	La Crosse	673	43–49	−43	−21	−25	S— 5.6	16– 4
	Milwaukee	591	43– 2	−25	−12	−15	W— 12.0	16– 5
Wyo.	Lander	5,370	42–50	−40	−12	−20	SW— 3.9	85– 9
	Sheridan	3,724	44–48	−45	−26	−25	NW— 5.1	48– 8
Alberta	Edmonton	2,183	53–30	−57	−41.3	−40	SW— 7.5	—
B. C.	Vancouver	38	49–18	2	13.1	15	E— 4.5	—
	Victoria	56	48–30	− 2	19.4	15	N— 12.6	—
Man.	Winnipeg	760	49–55	−54	−37.7	−40	NW—10.1	—
N. B.	Fredericton	31	45–57	−35	−25	−25	NW— 9.1	—
N. S.	Yarmouth	31	43–50	−12	− 0.1	0	NW—14.3	—
Ont.	London	804	42–59	−27	−13.7	−10	W— 10.3	—
	Ottawa	214	45–25	−35	−24.1	−20	W— 8.4	—
	Port Arthur	616	48–32	−40	−29.5	−30	NW— 8	—
	Toronto	254	43–40	−26	−11.2	−10	SW—13.6	—
P. E. I.	Charlottetown	9	46–16	−27	−13.2	−15	NW— 9.8	—
Que.	Montreal	63	45–30	−29	−18.4	−15	SW—11.3	—
	Quebec	21	46–50	−34	−23.2	−20	SW—13.3	—
Sask.	Prince Albert	1,414	53–15	−70	−47.3	−45	W— 5.1	—
Yukon	Dawson		64	−68	−54.3	−50	S— 3.7	—

* 1st Fig—Number of days in 23-yr period when daily minimum temp was equal to or lower than recommended design temp.
2nd Fig—Maximum number of days in one heating season, when daily minimum temp was equal to or lower than recommended design temp.

In most cases, this is correlated with the frequency of occurrence of the listed temperature as shown in Column G, which is explained later. It is interesting to note that there is some correlation between average annual minimum temperature and recommended design temperature. Some engineers prefer to use the former for design conditions.

Prevailing winter wind direction and average wind velocity in miles per hour are shown in column F, Table 1.

Column G gives design temperature frequency of occurrence. In this column, two figures are shown. The first is the number of days in a 23-year period when the daily minimum temperatures have been equal to or lower than the recommended design temperature shown in column E. The second figure is the maximum number of times during any one heating season that the daily minimum temperatures have been equal to or lower than the temperature in column E.

The design temperatures recommended show a frequency of occurrence in any one heating season of from two to fourteen times. The design temperature is attained more frequently in the upper central United States. In general, the recommended design temperatures occur about 15 to 40 times in a 23-year period and 5 to 10 times in any one heating season of that period. A comparison of this type permits local designers to adjust their design figures upward or downward, depending on frequency, local experience and type of construction.

Summer Weather Data

Tabular data of maximum recorded temperatures, recommended design dry bulb and wet bulb temperatures, prevailing summer wind direction and velocity in miles per hour are presented in **Table 2**.

The necessity for ventilation and control of both temperature and humidity requires knowledge of both dry and wet bulb design temperatures. In winter it is not customary to use the lowest temperature on record for design purpose, and, similarly in summer, the maximum outside dry bulb or wet bulb temperatures are not commonly considered for design.

The question of what summer design temperatures should be used is less controversial than that of winter design conditions. It has been almost universally accepted that the summer design outside dry-bulb temperature for any locality should be a temperature which is exceeded only 2.5 to 5% of the time during normal daylight hours in the summer season. Accepted design wet bulb temperatures are usually set at a figure which will be exceeded only 5% of the total hours during June to September inclusive. Strict application of these rules for establishing the design temperatures for any locality will result in temperatures expressed in degrees and tenths. It is customary for convenience to round out the figures for wet bulb temperatures to the nearest degree. Dry bulb temperatures are rounded out to some convenient figure close to and usually slightly higher than the result obtained by the 5% rule.

The maximum dry bulb and maximum wet bulb temperatures seldom occur simultaneously, and the use of off-peak conditions results in loads which are somewhat compensated for by this fact.

Table 2, column A, shows the highest dry bulb on record for representative cities in the United States. Recommended design outside dry bulb and wet bulb temperatures are listed in columns B and C, Table 2. Column D shows the prevailing summer wind direction and velocity in miles per hour.

The section of the table listing recommended design temperatures will be found to be consistent with the method previously outlined for establishing these conditions. In most cases the design dry bulb is rounded out to be within the 2.5 to 5% rule.

In the previous edition of the *Refrigerating Data Book*, in the chapter on Load Calculations, some recommended figures were offered as a guide to ambient temperatures. These were not intended to be used for air conditioning design, but were proposed for use in selection of air cooled condensing units and for heat leakage of commercial refrigerators.

The maximum ambient temperature was defined as the average of the highest temperature of the hottest day of five consecutive years. These figures for various

Table 2. Summer Weather Conditions

		A	B	C	D
State	City	Highest dry bulb on record	Design dry bulb	Design wet bulb	Summer wind direction and velocity
Ala.	Birmingham	107	95	78	S— 5.2
	Mobile	103	95	80	SW— 8.6
Ariz.	Phoenix	118	105	76	W— 6.0
	Tucson	111	105	72	W— 5.0
Ark.	Little Rock	110	95	78	NW— 6.0
Calif.	Los Angeles	109	90	70	SW— 6.0
	San Francisco	101	90	65	SW—11.0
Colo.	Denver	105	95	64	S— 6.8
Conn.	Hartford	101	93	75	S— 6.9
	New Haven	101	95	75	S— 7.3
Del.	Dover	104	93	78	SW— 9.0
	Wilmington	107	95	78	SW— 9.7
D. C.	Washington	106	95	78	S— 6.2
Fla.	Jacksonville	104	95	79	SW— 8.7
	Miami	96	91	79	SE— 8.2
	Tampa	98	94	79	E— 7.0
Ga.	Atlanta	103	95	76	NW— 7.3
	Savannah	105	95	78	SW— 7.8
Idaho	Boise	121	95	65	NW— 5.8
Ill.	Chicago	105	95	75	NE—10.2
	Peoria	113	95	76	S 8.2
Ind.	Fort Wayne	106	95	75	SW— 8.0
	Indianapolis	106	95	76	SW— 9.0
Iowa	Des Moines	110	95	77	SW 6.6
	Sioux City	111	96	75	S— 10.1
Kan.	Topeka	114	102	76	S— 10.0
	Wichita	114	100	75	S— 11.0
Ky.	Louisville	107	95	76	SW— 8.0
La.	New Orleans	102	95	79	SW— 7.0
	Shreveport	110	100	78	S— 6.2
Me.	Portland	103	90	73	S— 7.3
Md.	Baltimore	107	95	78	SW— 6.9
Mass.	Boston	104	92	75	SW— 9.2
	Springfield	104	93	75	SW— 9.0
Mich.	Detroit	105	95	75	SW—10.3
	Grand Rapids	108	95	75	W— 7.7
Minn.	Duluth	106	93	73	NE— 10.2
	Minneapolis	108	95	75	SE— 8.4
Miss.	Vicksburg	104	95	78	SW— 6.2
Mo.	Kansas City	113	100	76	S— 9.5
	St. Louis	110	95	78	SW— 9.4
Mont.	Helena	103	95	67	SW— 7.3
Neb.	Omaha	114	95	77	S— 7.7
Nev.	Reno	106	95	65	W— 7.4
N. H.	Concord	102	87	72	NW— 5.4
N. J.	Atlantic City	104	92	77	SW—13.4
	Newark	105	95	75	SW—12.6
	Trenton	106	95	78	SW—10.0
N. Mex.	Santa Fe	97	90	65	SE— 6.5
N. Y.	Albany	104	92	75	S— 7.1
	Buffalo	97	93	75	SW—12.2
	New York	102	95	75	SW—12.9
N. C.	Asheville	97	90	75	SE— 5.6
	Charlotte	103	94	75	SW— 4.8
	Raleigh	103	94	78	SW— 5.9
N. D.	Bismarck	114	95	73	NW— 8.8
Ohio	Cincinnati	108	95	78	SW— 6.6
	Cleveland	100	95	75	S— 9.9
	Columbus	106	95	76	SW— 8.5
	Toledo	105	95	75	SW— 9.7
Okla.	Okla. City	113	101	76	S— 10.1
	Tulsa	115	101	77	S— 10.0
Ore.	Portland	105	90	65	NW— 6.6
Penna.	Philadelphia	106	95	78	SW— 9.7
	Pittsburgh	103	95	75	NW— 9.0
	Scranton	103	95	75	SW— 5.8
R. I.	Providence	100	93	75	NW—10.0
S. C.	Charleston	104	95	80	SW— 9.9
	Greenville	106	95	76	NE— 6.8
S. D.	Sioux Falls	110	95	75	S— 7.6

Table 2. Summer Weather Conditions (Concluded)

State	City	A	B	C	D
		Highest dry bulb on record	Design dry bulb	Design wet bulb	Summer wind direction and velocity
Tenn.	Chattanooga	104	95	77	SW— 6.5
	Knoxville	104	95	78	SW— 5.9
	Memphis	106	95	78	SW— 7.5
	Nashville	106	95	78	W— 7.5
Tex.	Dallas	110	100	78	S— 9.4
	El Paso	106	100	69	E— 6.9
	Galveston	101	95	80	S— 9.7
	Houston	108	95	78	S— 7.7
	San Antonio	107	100	78	SE— 7.4
Utah	Salt Lake City	105	92	63	SE— 8.2
Vt.	Burlington	100	90	73	S— 8.9
Va.	Norfolk	105	95	78	S— 10.9
	Richmond	107	95	78	SW— 6.2
	Roanoke	105	95	76	
Wash.	Seattle	98	85	65	S— 7.9
	Spokane	108	90	65	SW— 6.5
W. Va.	Charleston	108	95	75	SW— 4.0
Wis.	Milwaukee	105	95	75	S— 10.4
Wyo.	Cheyenne	100	95	65	S— 9.2

cities, when compared with the recommended design dry bulb temperatures for air conditioning, revealed that the maximum ambient temperature ranged from seven degrees below the design dry bulb to as much as nine degrees above the design dry bulb temperature. It is recommended that five degrees above be used for general practice, with normal ventilation. For poorly ventilated spaces where pumps, compressors, and motors are operating, the ambient temperature in the space may rise to twenty degrees or more above the normal outside ambient temperature, and in extreme cases, the more conservative figures should be used.

Summer Water Temperature Data

Water temperature data from various sources for key cities and states are shown in **Table 3**. Most of these data are from tables some of which have been previously published but all of which originated in reports published by the United States Department of Commerce.

Water temperatures from mechanical draft cooling towers are dependent upon design wet bulb temperatures and cooling tower efficiencies. Information on cooling tower design may be found in Chapter 21. The data on water temperatures in this chapter are limited to those of water from natural sources, either surface or well. In most cases, city water mains are supplied from one of these two sources, and it is unusual to have both sources in one locality except when private wells are used.

These water temperatures are based on the highest monthly average occurring in any of the four months, June to September inclusive, and represent only one or two years' record. Most of these data were obtained and compiled originally in the period from 1930 to 1935, but past experience has shown that they do not change to any great degree from year to year. For application of self-contained air-conditioning units or commercial condensing units, cooled with city water, the temperatures listed in Table 3 will be satisfactory for purposes of estimating.

In any locality, application engineers concerned with local conditions may check local weather bureau records for information on dry and wet bulb temperatures. Water temperature data may be obtained from municipal engineers' offices, utility company or industrial plant records.

For other weather data not shown in the tables in this chapter, various sources are recommended. The United States Yearbook of Agriculture, 1941 Edition, contains information on maximum and minimum dry bulb temperatures, rainfall statistics and dates of first and last killing frosts. Probably one of the most complete summaries of weather information useful in design of summer air conditioning equipment is "Summer Weather Data" published by the Marley Company, Kansas

Table 3. City Water Temperatures

City	Water source*	Water temp, F	City	Water source*	Water temp, F
Abilene, Tex.	S-W	80.0	Chicago, Ill.	S	69.4
Akron, Ohio	S	74.7	Chicago Heights, Ill.	W	55.0
Alameda, Calif.	S	64.0	Chicopee, Mass.	S	66.7
Albany, N.Y.	S	66.0	Cicero, Ill.	S	72.0
Albuquerque, N.M.	W	72.0	Cincinnati, Ohio	S	82.0
Alexandria, La.	W	86.0	Clarksburg, W.Va.	S	77.0
Alexandria, Va.	S	74.0	Cleveland, Ohio	S	73.5
Alhambra, Calif.	W	68.0	Clifton, N.J.	S	55.0
Allentown, Pa.	S-W	70.0	Colorado Springs, Colo.	S	62.0
Alliance, Ohio	S-W	68.0	Columbus, Ga.	S	80.0
Amarillo, Tex.	W	65.0	Columbus, Ohio	S	76.0
Ambridge, Pa.	W	66.0	Concord, N.H.	S-W	73.0
Amsterdam, N.Y.	S	57.0	Council Bluffs, Ia.	S	81.5
Anderson, Ind.	S-W	87.0	Covington, Ky.	S-W	82.0
Ann Arbor, Mich.	W	62.5	Cranston, R.I.	—	65.0
Anniston, Ala.	W	67.0	Cumberland, Md.	S	65.0
Appleton, Wis.	S	76.0	Dallas, Tex.	S-W	82.5
Asheville, N.C.	S	76.0	Danville, Ill.	S	84.0
Ashland, Ky.	S	82.4	Davenport, Ia.	S	83.9
Atlanta, Ga.	S	83.5	Dearborn, Mich.	—	75.0
Atlantic City, N.J.	S-W	72.5	Decatur, Ill.	S	83.3
Attleboro, Mass.	S-W	68.7	Denver, Colo.	S-W	70.9
Auburn, N.Y.	S	69.0	Des Moines, Ia.	S	72.9
Augusta, Ga.	S	85.0	Detroit, Mich.	S	75.0
Aurora, Ill.	W	60.0	Dubuque, Ia.	W	60.0
Austin, Tex.	S	85.0	Duluth, Minn.	S	70.6
Baltimore, Md.	S	70.0	Durham, N.C.	S	76.2
Bangor, Me.	S	72.0	East Cleveland, Ohio	—	73.5
Battle Creek, Mich.	S-W	52.0	East Providence, R.I.	S	60.0
Bay City, Mich.	S	78.0	East St. Louis, Ill.	S	90.0
Beaumont, Tex.	S	88.0	Easton, Pa.	S	65.0
Belleville, Ill.	S	90.0	Eau Claire, Wis.	—	53.0
Beloit, Wis.	W	57.0	Elgin, Ill.	—	56.0
Berkeley, Calif.	S	64.0	Elizabeth, N.J.	S-W	66.7
Berlin, N.H.	S	66.0	Elkhart, Ind.	W	60.0
Bethlehem, Pa.	S-W	75.0	El Paso, Tex.	W	85.0
Birmingham, Ala.	S	82.0	Elyria, Ohio	S	72.9
Bloomington, Ill.	S	78.8	Erie, Pa.	S	72.9
Bluefield, W.Va.	S	70.6	Evanston, Ill.	S	69.0
Boston, Mass.	S	74.3	Evansville, Ind.	S	85.0
Braddock, Pa.	S	80.0	Everett, Wash.	S	62.0
Bradford, Pa.	S-W	60.0	Fargo, N.D.	S	85.1
Bridgeport, Conn.	S	64.0	Findlay, Ohio	S-W	63.0
Brockton, Mass.	S	62.0	Flint, Mich.	S	72.0
Brookline, Mass.	W	67.0	Fond du Lac, Wis.	W	56.0
Brownsville, Tex.	S	87.0	Fort Dodge, Ia.	W	52.0
Buffalo, N.Y.	S	73.0	Fort Smith, Ark.	S	80.0
Burlington, Ia.	S	87.0	Fort Wayne, Ind.	S	79.0
Burlington, Vt.	S	66.0	Fort Worth, Tex.	S	83.0
Butler, Pa.	S	70.0	Freeport, Ill.	W	56.0
Cambridge, Mass.	S	74.0	Fresno, Calif.	W	72.0
Camden, N.J.	W	58.0	Gadsden, Ala.	S	82.0
Carbondale, Pa.	S	70.0	Galesburg, Ill.	W	67.0
Cedar Rapids, Ia.	S	82.5	Gary, Ind.	S	70.0
Central Falls, R.I.	—	74.0	Glendale, Calif.	S-W	68.2
Charleston, S.C.	S	85.0	Grand Rapids, Mich.	S	74.0
Charlestown, W.Va.	S	83.0	Granite City, Ill.	—	90.0
Charlotte, N.C.	S	81.0	Greenville, S.C.	S	76.0
Chattanooga, Tenn.	S	79.5	Hagerstown, Md.	S	83.0
Chester, Pa.	S	79.0	Hamilton, Ohio	W	65.0

* S = surface; W = well.

Table 3. City Water Temperatures (Continued)

City	Water source*	Water temp, F	City	Water source*	Water temp, F
Hammond, Ind.	S	82.0	Newark, N.J.	S	71.5
Hannibal, Mo.	S	80.0	Newark, Ohio	S	70.0
Hartford, Conn.	S	70.0	New Bedford, Mass.	S	71.0
Hazelton, Pa.	S	65.0	New Brunswick, N.J.	S	71.0
Highland Park, Mich.	S	74.0	Newburgh, N.Y.	S	54.0
High Point, N.C.	S	78.0	New Haven, Conn.	S	70.0
Holyoke, Mass.	S	74.0	New Orleans, La.	S	90.0
Houston, Tex.	W	84.0	Newport, R.I.	S	70.2
Hutchinson, Kan.	W	60.0	Newport News, Va.	S-W	82.0
Indianapolis, Ind.	S-W	82.0	New York, N.Y.	S-W	70.0
Ithaca, N.Y.	S	71.6	Niagara Falls, N.Y.	S	69.8
Jackson, Mich.	W	52.0	Norfolk, Va.	S	83.0
Jackson, Miss.	S	82.0	Norristown, Pa.	S	84.0
Jackson, Tenn.	W	70.0	North Little Rock, Ark.	S	85.0
Jacksonville, Fla.	W	86.7	North Tonawanda, N.Y.	S	78.0
Janesville, Wis.	W	52.0	Norwood, Ohio	W	56.0
Jefferson City, Mo.	S	80.0	Oakland, Calif.	S	64.0
Johnstown, Pa.	S	59.5	Oak Park, Ill.	S	75.0
Kalamazoo, Mich.	W	52.0	Ogden, Utah	S-W	60.0
Kansas City, Kan.	S	93.0	Oil City, Pa.	W	65.0
Kenosha, Wis.	S	67.0	Oklahoma City, Okla.	S	77.2
Knoxville, Tenn.	S	84.3	Omaha, Nebr.	S	80.9
LaCrosse, Wis.	W	52.0	Orlando, Fla.	S	87.0
Lafayette, Ind.	W	53.0	Oswego, N.Y.	S	58.0
Lancaster, Pa.	S	77.0	Ottumwa, Ia.	S	77.5
Lansing, Mich.	W	59.0	Paducah, Ky.	S	89.0
Laredo, Tex.	S	84.6	Pasadena, Calif.	S-W	74.0
Leominster, Mass.	S	74.0	Passaic, N.J.	S	55.0
Lima, Ohio	S	77.4	Paterson, N.J.	S	64.0
Lincoln, Nebr.	W	59.0	Pawtucket, R.I.	S	74.0
Little Rock, Ark.	S-W	88.0	Pensacola, Fla.	W	70.0
Los Angeles, Calif.	S-W	76.0	Peoria, Ill.	W	56.0
Louisville, Ky.	S	82.0	Perth Amboy, N.J.	W	54.0
Lowell, Mass.	W	50.0	Petersburg, Va.	S	78.0
Lubbock, Tex.	W	67.0	Phoenix, Ariz.	S	81.0
Lynchburg, Va.	S	73.0	Philadelphia, Pa.	S	79.0
McKeesport, Pa.	S	80.6	Pine Bluff, Ark.	W	80.0
Madison, Wis.	W	53.0	Pittsburgh, Pa.	S	80.6
Manitowoc, Wis.	W	58.0	Pontiac, Mich.	W	55.0
Mansfield, Ohio	W	50.0	Portland, Me.	S	66.0
Marion, Ind.	W	55.0	Portland, Ore.	S	62.0
Mason City, Ia.	W	59.0	Portsmouth, Ohio	S	80.6
Massillon, Ohio	S-W	55.0	Poughkeepsie, N.Y.	S-W	76.6
Maywood, Ill.	W	60.0	Providence, R.I.	S	65.0
Memphis, Tenn.	—	70.0	Pueblo, Colo.	S	74.0
Meridian, Miss.	S-W	86.0	Quincy, Ill.	S	82.2
Michigan City, Ind.	S	71.0	Quincy, Mass.	S	74.0
Milwaukee, Wis.	S	60.6	Racine, Wis.	S	68.5
Minneapolis, Minn.	S	80.2	Raleigh, N.C.	S	83.4
Mishawaka, Ind.	W	57.0	Reading, Pa.	S	78.8
Mobile, Ala.	—	73.0	Richmond, Va.	S	80.4
Moline, Ill.	S	85.9	Riverside, Calif.	W	74.0
Montgomery, Ala.	W	71.0	Rochester, Minn.	W	58.0
Mount Vernon, N.Y.	S	68.0	Rochester, N.Y.	S	68.9
Muncie, Ind.	S-W	75.5	Rockford, Ill.	W	55.0
Muskegon, Mich.	S	68.0	Rock Island, Ill.	S	80.0
Muskogee, Okla.	S	90.7	Rome, Ga.	S	78.0
Nashua, N. H.	W	67.0	Sacramento, Calif.	S	80.6
Nashville, Tenn.	S	88.0	Saginaw, Mich.	S	78.0
New Albany, Ind.	S	83.0	St. Cloud, Minn.	S	77.0

* S = surface; W = well.

Table 3. City Water Temperatures (Concluded)

City	Water source*	Water temp, F	City	Water source*	Water temp, F	Design wet bulb, F
St. Louis, Mo.	S	85.0	Texarkana, Tex.	S-W		86.0
St. Louis County, Mo.	S	84.0	Toledo, Ohio	S		84.0
Salina, Kan.	W	62.0	Topeka, Kan.	S-W		86.0
Salt Lake City, Utah	S-W	58.0	Trenton, N.J.	S		79.0
San Angelo, Tex.	S	80.0	Troy, N.Y.	S		72.5
San Antonio, Tex.	W	76.0	Tucson, Ariz.	W		80.0
Sandusky, Ohio	S	75.2	Tulsa, Okla.	S		81.8
San Francisco, Calif.	S-W	60.0	Tuscaloosa, Ala.	S		71.0
San Jose, Calif.	S-W	73.0	Uniontown, Pa.	S-W		68.0
Santa Ana, Calif.	W	69.0	Utica, N.Y.	S		70.2
Santa Barbara, Calif.	S	70.0	Waco, Tex.	S		84.6
Schenectady, N.Y.	W	57.0	Warren, Ohio	S		78.6
Scranton, Pa.	S	70.9	Washington, D.C.	S		75.0
Seattle, Wash.	S	62.0	Waterbury, Conn.	S		74.0
Shamokin, Pa.	S	65.0	Watertown, N.Y.	S		72.0
Sharon, Pa.	S	73.0	Waukegan, Ill.	S		69.8
Shawnee, Okla.	S	55.0	Wausau, Wis.	W		60.0
Sheboygan, Wis.	S	65.0	Westfield, Mass.	S		63.0
Shreveport, La.	S	91.0	West Palm Beach, Fla.	S		86.2
Sioux Falls, S.D.	W	55.0	Weymouth, Mass.	S		75.2
Spartanburg, S.C.	S	78.8	Wheeling, W.Va.	S		80.9
Spokane, Wash.	S	47.0	Wichita Falls, Tex.	S		90.0
Springfield, Ill.	S	80.6	Wilkinsburg, Pa.	S		79.0
Springfield, Mass.	S	55.0	Williamsport, Pa.	S-W		66.0
Springfield, Mo.	S-W	76.0	Wilmington, Del.	S		79.0
Stamford, Conn.	S	76.0	Wilmington, N.C.	S		86.0
Steubenville, Ohio	S	68.0	Winona, Minn.	W		52.0
Stockton, Calif.	W	70.0	Woodbridge, N.J.	S		51.8
Stratford, Conn.	S	64.0	Woonsocket, R.I.	S		72.0
Superior, Wis.	W	64.4	Worcester, Mass.	S		73.0
Syracuse, N.Y.	S	70.4	Yakima, Wash.	S		65.0
Tacoma, Wash.	S-W	61.0	Yonkers, N.Y.	S		79.0
Tampa, Fla.	S	87.0	Youngstown, Ohio	S		68.5
Taunton, Mass.	S	70.0				

* S = surface; W = well.

City, Kan. This book contains analytical data on a nationwide basis of dry bulb temperatures, wet bulb temperatures, dew points, wind and sunshine for the summer period of June to September inclusive, and much information on conditions at other times of the year. Material in this book has been drawn upon freely in the preparation of this chapter.

The frequency of occurrence of recommended design temperatures for winter is based upon a study, the results of which were published in 1939 by the Carnegie Institute of Technology—Bulletin No. 82, "An Analysis of Winter Temperatures for One Hundred and Twenty Cities," by Clark M. Humphreys.

Summer water temperature data are from several sources, including a table appearing in earlier editions of the *Refrigerating Data Book*, tables in "Heating and Ventilating Data Sheets" which were originally published by the Department of Commerce, and a few local sources.

12. PHYSICAL PROPERTIES OF MATERIALS, INCLUDING INSULATION

Table 1. Gases and Vapors

Name or description	Formula	Sp ht C_p		Density at 1 atm and 32F lb/cu ft	Mol wt	Boiling point F	Thermal conductivity		Critical temp F	Pressure psia
		Btu/lb	Temp, F				F	K		
Air		.2377	−22 to +50	.0808	28.97	−317	−148 32	.0095 .0140	221	910
Air		.2375	32 to 400				212	.0183		
Alcohol	C_2H_5OH	.4534	110 to 400							
Alcohol	CH_3OH	.4580	110 to 400							
Ammonia	NH_3	.5202	73 to 212	.0482	17.03	− 28	− 76 + 32	.0095 .0128	270	1,638
Argon	A	.1233	68 to 195	.1114	39.91	−302			−187	705
Acetylene	C_2H_2			.0732	26.02					
Benzene	C_6H_6	.2990	94 to 235	.206	78.11	176.18	32	.0052	551	700
Bromine	Br_2	.0555	181.4 to 442.4	.4460	159.83					
Butane	C_4H_{10}			.1623	58.08	31.1	32	.0078	307	528
Benzol				.233						
Butylene	C_4H_8			.148	56.06					
Carbon dioxide	CO_2	.1843	−20 to 45			−164.2	− 58 32	.0068 .0085		
Carbon dioxide	CO_2	.2025	59 to 212	.1235	44.0				88	1,072
Carbon monoxide	CO	.2425	74 to 210	.0781	28.0	−311.5	32	.0135	−218	514
Chlorine	Cl_2	.1125	61 to 700	.2011	70.91	− 22	32	.0043	291	1,118
Chloroform	$CHCl_3$	.1441	80 to 230		119.39	142.16	32	.0038		
Carbon disulfide	CS_2				76.1	115			523	1,116
Carbon tetrachloride	CCl_4				153.8	170			541	661
Dichlorodifluoromethane (F_{12})	CCl_2F_2	.143	0 to 100		120.92	−21.64	32	.0048		
Dichloromethane	CH_2Cl_2				84.9	105			421	1,490
Ether	$C_4H_{10}O$	.4797	156.2 to 435.2		74.12	34	− 94	.0066	90	717
Ethane	C_2H_6			.0848	30.05	−126.94	− 29 32	.0086 .0106		
Ethyl chloride	C_2H_5Cl			.1793	64.5	54.5	32	.0055	370	764
Ethylene	C_2H_4			.0783	28.03	−155	− 96 32	.0064 .0101	50	747
Ethyl ether	$C_4H_{10}O$	.4280	77 to 232				32	.0077		
Fluorine	F_2			.1022	38.00	−304.6				
Freon-12	CCl_2F_2	.143	1 to 100		120.9	− 21	207	.057	233	580
Helium	He			.011	4.00	−452			−450	33
Hydrochloric acid	HCl	.1940	55.4 to 212	.1024	36.47					
Hydrogen	H_2	3.3996	−2.2 to +118.4	.0056	2.016	−421.6	− 58 32	.083 .100	−400	188
Hydrogen sulfide	H_2S	.2451	68 to 402.8	.096	34.08	139.28	32	.0076	212	1,306
Hydrogen chloride	HCl				36.4	−121			124	1,198
Heptane	C_7H_{16}			.301	100.2	209.08	392	.0112		
Hexane	C_6H_{14}			.2553	86.17	155.66	32	.0072	454	433
Hexylene	C_6H_{12}				84.09					
Isobutane	C_4H_{10}			.1669	58.07	14	32	.0080	273	543
Isopentane	C_5H_{12}				72.09					
Methyl chloride	CH_3Cl			.1440	50.48	− 10.66	32	.0053	289	966
Methane	CH_4	.5929	66 to 390	.0448	16.03	−257.8			−116	672
Neon	Ne			.0544	20.40	−320.44			−380	389
Nitrogen		.2438	32 to 390				32	.0140		
Nitrous oxide	N_2O	.2126	80 to 220	.1235	44.02	−129.1	− 98 32	.0067 .0087	98	1,053
Nitrogen tetroxide	NO_2	1.625	80 to 150							
Naphthalene	$C_{10}H_8$				128.0					
Natural gas					19.46					
Oxygen	O_2	.2175	50 to 400	.0892	32.00	−297.4	− 58 32	.0119 .0142	−182	730
Propane	C_3H_8			.126	44.09	− 44.14	32	.0087	204	632
Pentane	C_5H_{12}			.211	72.15	96.8	32	.0074	387	485
Phenol	C_6H_6				94.0					
Propylene	C_3H_6				42.0	− 52			198	661
Refinery gas					1.20					
Sulfur dioxide	SO_2	.1544	16 to 202	.1828	64.06	+ 14	32	.0050	315	1,141
Water vapor	H_2O	.46 .49	100 210		18.01	212	115 212	.0120 .0137	706	3,226

Note: *K* = Btu ft/hr sq ft F, however lower case *k* is *k* = Btu in./hr sq ft F.

Table 2. Liquids

Name or description	Formula	Boiling point F	Enthalpy of vaporization Btu/lb	Sp ht		Viscosity		Freezing point F	Enthalpy of fusion Btu/lb	Sp gr or density (ρ)		Thermal conductivity, K		Vapor pressure	
				Btu/lb	Temp F	Centipoises	Temp F			Temp F	lb/cu ft ρ	Btu ft per hr sq ft F	Temp F	mm of Hg	Temp F
Acetaldehyde	C_2H_4O	69.44	244.8			.275	32	−190.3						400	42.8
Acetic acid	$C_2H_4O_2$	245.3	173.	.522	78.8–203	1.040	86	62.00	77.7			.099	68	200	176
Acetone	C_3H_6O	132.98	223.	.506	32			203	42.1	68	49.4 (ρ)	.102	77–86	185	68
Allyl alcohol	C_3H_6O	206.6	293.	.665	69.8–204.8	1.168	86	−200.2				.104	77–86		
Amyl alcohol	$C_5H_{12}O$	280.22	216.					−109.3	48.			.094	86	100	177.8
Ammonia	NH_3			1.099	32				94.5			.29	5–86		
Alcohol-ethyl	C_2H_6O	172.94	366.9	.548	0	1.2	68	−174.28	45.6			.789		44	68
Alcohol-methyl	CH_4O	148.46	472.	.601	64	.596	68	−142.6	39.6			.796		94	68
Aniline	$C_6H_5NH_2$	363.2	198.	.514	60	4.467	68	+ 20.77	48.6	32	64.5 (ρ)			10	156.2
Benzol	C_6H_6			.340	50	.567	86		55.						
Benzene	C_6H_6	176.18	167.						54.6	32	56.1 (ρ)	.092	86	75	68
Bromine	Br	137.84	86.4	.107	13–45	.911	86		29.15	32					
Butyl-alcohol	$C_4H_{10}O$	243.86	254.	.687	20–115			−129.64	54.			.097	86		
Butyric acid	$C_4H_8O_2$	326.3	205.	.515	20–100	1.304	86	+ 22.01	54.2					50	197.6
Calcium chloride brine	$CaCl_2$			.764	5					5	1.14				
				.787	68					68	1.14	30% .32	86		
				.695	+4					+4	1.20	15% .34	86		
				.725	68					68	1.20				
				.651	−4					−4	1.26				
				.676	68					68	1.26				
Carbolic acid	C_6H_6O	359.96		.561	57.2–78.8				52.2	59	60.2 (ρ)				
Carbon disulfide	CS_2	115.27	151.3	.240	68	.352	86	−169.24		32	80.6 (ρ)	.093	86	100	+23
Carbon tetrachloride	CCl_4	169.16	83.5	.201	68	.848	86	− 9.04	74.8			.107	32	90	68
Chloroform	$CHCl_3$	142.16	106.	.226	59	.519	86	− 82.3			1.50	.080	86	160	68
Copper sulfate	$CuSO_4$														
	$+ 50\ H_2O$			.848	58										
	$+100\ H_2O$			.951	57										
	$+400\ H_2O$			.975	59										
Diphenylamine	$C_{12}H_{11}N$	575.6		.464	125			127.36						10	314.6
Decane	$C_{10}H_{22}$	345.2	108.2	.500	0–50	.77	72.14	− 22	86.4			.085	86	10	132.8
Dow Corning 500		211.1 305.6 377.6 446.0													
Ethyl ether	$C_4H_{10}O$	94.08	159.	.529	32	.223	86	−176.8	41.4		.736	.08	86	442	68
Ethyl acetate	$C_4H_8O_2$	170.78	183.5 (32 F)	.475	68			−118.84	51.1			.101	68	72.8	68
Ethyl alcohol	C_2H_6O	172.94	367.	.548	32				44.8			.105	68	50	72.86
Ethyl bromide	C_2H_5Br	101.12	108.	.215	59–68	.368	86	−182.2						386	68
Ethyl chloride	C_2H_5Cl	53.96	166.5	.367	32			−217.66						1,000	68
Ethyl iodide	C_2H_5I	161.78	81.3	.161	68	.540	86	−163.3				.064	104	108	68
Ethylene bromide	$C_2H_4Br_2$	259.06	83.	.173	68	1.475	86	50.108						10	72.86
Ethylene chloride	$C_2H_4Cl_2$	182.66	139.	.299	68	.736	86	− 31						100	85.46
Ethylene glycol	$C_2H_6O_2$	386.6	344.									.153	32	10	192.56
Formic acid	CH_2O_2	213.44	216.	.525	68–212	1.46	86	47.12	104.4					50	84.2
Glycerine	$C_3H_8O_3$	554		.575	59–120	830.	68.54	64.58	85.6					385	−184
Glycerol	$C_3H_8O_3$	555.08				207.	68	64.4	85.5			.164	68		

Gasoline		158–194		.5	32–212						.73				
Heptane	C_7H_{16}	209.12	137.1	.490	68	.375	86	−131.08	60.6			.081	86	50	80.96
Hexane	C_6H_{14}	755.6	142.5	.600	68	.296	86	−139	65.			.080	86	100	61.16
Hydrochloric acid	HCl													100	140.18
Isobutyl alcohol	$C_4H_{10}O$	224.42	248.												
Kerosene				.5	32–212						.78–.82	.086	68		
Linseed oil						33.1	86				.925				
Methyl acetate	$C_4H_6O_2$	134.78	176.5	.468	59			−144.49						200	74.84
Methyl iodide	CH_3I	108.14	82.6			.460	86	− 86.98						400	76.64
Naphthalene	$C_{10}H_8$	424.4	136.	.396	185			176.396	64.					10	176
Nitrobenzole				.350	58										
Nitric acid	HNO_3	186.8						− 43.6	17.15		91 %—1.50				
Nitrobenzene	$C_6H_5NO_2$	411.62	142.2	.350	57.2			42.53	40.5			.095	86	10	185.9
Nonane	$CH_3(CH_2)_7CH_3$	302		.503	32–122	.62	72.14	− 64.66				.084	86	100	199.94
Oils															
Castor				.434		451.	86			59	60.5 (ρ)	.104	68		
Citron				.438	42										
Olive				.471	44	54.	86			59	.906	.097	68		
Sesame				.387											
Rapeseed						42.2	100.04			59	57. (ρ)				
Soybean						40.6	86			194	.919				
Sperm						42.	60.08			77	55. (ρ)				
Octane	$CH_3(CH_2)_6CH_3$	256.28	127.5	.587	68–253.4	.483	86	− 70.42				.083	86	10	66.2
Petroleum				.511	70–135						.87				
Pentane	C_5H_{12}	96.8				.220	86	−201.82		32	40.6 (ρ)			400	65.84
Propionic acid	$C_3H_6O_2$	285.98	177.8	.560	68–278.6	.960	86	− 5.44						10	104
Propyl alcohol	C_3H_8O	207.5	295.2	.57	68	1.779	86	−194.98			2.04			10	57.56
Potassium hydroxide	2 KOH														
	$KOH + 30\ H_2O$			.876	64										
	$KOH + 100\ H_2O$			.975	64										
Sea water				.980							1.004				
				.938							1.024				
				.903							1.046				
Sulfuric acid 100 %	H_2SO_4	626	219.6	.344	68			50.882	43.2		87 %—1.80	.21	86		
Sodium chloride brine	$NaCl$														
	$+ 10\ H_2O$			.791	64						25 %—.33		86		
	$+200\ H_2O$			.978	64						13 %—.34		86		
Sodium hydroxide	$NaOH + 50\ H_2O$			.942	64										
	$NaOH + 100\ H_2O$			.983	64										
Toluol	C_6H_8	231.8	154.8	.364	50										
				.534	185										
Toluene	C_7H_8	230.54	155.7	.440	53.6–210.2	.525	86	−139							
Turpentine	$C_{10}H_{16}$	320	123.4	.411	32	1.272	86				.864	.074	59	.4	68
Water	H_2O	212	969.7	1.000	60.8	.8007	86	32	143.05	39.2	1.00	.330	32	10	52.358
										39.2	62.4 (ρ)	.356	86	100	125.06
Xylene	$C_6H_4(CH_3)_2$	287.6	149.2	.411	86			− 16.78				.090	68		
Zinc sulfate	$ZNSO_4$														
	$+ 50\ H_2O$			.842	68 to 125										
	$+200\ H_2O$			.952	125										

Table 3. Solids

Name or description	Sp ht		Sp gr	Conductivity Btu ft/hr sq ft F		Emissivity	
	Btu/lb F	Temp F		Temp F	*K*	Temp F	Ratio
Aluminum	.226	100	2.55–2.80	64	116.0	73	.040
Aluminum bronze			7.7				
Alundum	.186	212					
Asbestos	.25	.47–.58	2.1–2.8	32	.09	100–700	.93–.95
Asphalt	.3–.4						
Ashes	.20		.64–.72	32	.041		
Bakelite	.3–.4						
Brickwork	.2		1.85–2.00	70	.33–.92	70	.93
Brass, red	.08991	32	8.4–8.7	32	59.5	73	.043
Brass, yellow	.08831	32	8.4–8.7	32	49.4		.043
Bismuth tin	.040			64	37.6		
Bell metal	.086	59–208.4					
Bronze	.104		7.4–8.8				
Cadmium	.0548		8.65	64	53.7		
Carbon (gas retort)	.204					68	.927–.967
Cardboard					.1–.2		
Cellulose	.32						
Cement, Portland clinker	.186		1.5–2.4		.017		
Charcoal (wood)	.242		.28–.57	172	.051		
Chrome brick	.17						
Clay	.224		1.28				
Coal	.26–.37		.65–1.8				
Coal tars	.35	104					
Coal tar oils	.34	59–194					
Coke	.265	69.8–752	1.0–1.4	32	.106		
Concrete (stone)	.156	70–213	1.5–2.4		.5–.75		
Copper (cast rolled)			8.8–8.9	64	222.0	66	.030
Cryolite	.253	60.8–131					
Chalk	.215		1.8–2.8		.48	145–380	.32–.36
Cork (granulated rolled)	.485		.22–.26	24	.028		
Cotton (flax, hemp)			1.47–1.50	32	.033		
Cotton (wool)					.01		
Diamond	.147						
Earth (dry & packed)			1.5 loose	32	.035	145	.39–.43
				100	.039		
Felt				86	.022		
Fireclay brick	.198	212					
Fluorspar	.21	86					
Glass (crown)	.16–.2		2.4–2.7		.333–.5	72	.937
(flint)	.117		3.2–4.7			72	.937
(pyrex)	.20					72	.937
(silicate)	.188–.2	32–212				72	.937
(wool)	.157						
(common)			2.40–2.80			72	.937
Graphite (powder)	.165	78.8–168.8		104	.106		
Graphite	.20	68–212	2.4–2.7		1.0–2.32	145–380	.40–.56
Gypsum	.259	60.8–114.8	2.3–2.8	68	.25	70	.903
German silver	.0946	32–212	8.58				
Garnet	.1758	60.8–212					
Gold	.0308		19.25–19.2	64	169.0		.05
Ice	.350	−112	.88–.92	32	1.28 (water)	32	.95
Ice	.434	− 40		14	1.35		
Ice	.465	− 4		− 4	1.41		
Ice	.487	32		−22	1.471		
Ice				−40	1.538		
India rubber (para)	.481	−148					
Iron (gray cast)	.101		7.03–7.13	129	27.6		

Table 3. Solids (Concluded)

Name or description	Sp ht		Sp gr	Conductivity Btu ft/hr sq ft F		Emissivity	
	Btu/lb F	Temp F		Temp F	*K*	Temp F	Ratio
Iron (cast pig)			7.2				
Iron (wrought)			7.6–7.9	64	34.9	70–680	.94
Lead	.030		11.34	64	20.1	75	.281
Limestone	.217	59–212	2.1–2.8		.3–.75	145–380	.36–.90
Litharge	.055						
Leather			.86–1.02		.092		
Linen					.05		
Marble	.21	64.4	2.4–2.8		1.2–1.7	72	.931
Manganese			7.42				
Magnesia	.234	212			.04		
Magnesite brick	.222	212			.9–2.5		
Monel metal	.127	68–2372	8.97				
Mica	.10	68			.44		
Nickel	.103		8.9	64	34.4	74	.045
Nickel steel	.109						
Paper	.324		.70–1.15		.075	66	.924
Paraffin	.6939	32–68	.87–.91	86	.145		
Platinum (cast)			21.5	64	40.2		
Porcelain	.22			329	.945	72	.924
Pyrites (copper)	.131	66.2–122					
Pyrites (iron)	.136	59–208.4					
Paint (lacquers, varnish)							
White lacquer						100–200	.80–.95
White enamel						73	.906
Black lacquer						76	.875
Black shellac						70	.821
Aluminum paint						212	.3–.5
Plaster (rough lime)					.25–.05	50–190	.91
Sawdust			.21	68	.042		
Rock salt	.219	55.4–113					
Rubber (goods)	.48		1.0–2.0	100	.92	76	.859
Saltpeter			1.07				
Sand	.191		1.4–1.9	68	.188		
Silica	.316						
Steel (cold-drawn)	.12		7.83	64	26.2	75	.82
Stone	.2						
Silver (cast)			10.4–10.6	64	244.0	75	.82
Snow (fresh fallen)			.125				
Tin (cast)	.053		7.2–7.5	64	37.6	76	.043–.064
Tungsten	.034		19.22				
Tar (bituminous)			1.20				
Wood (oak)	.570		.65–.84		.085–.125	70	.895
most woods vary between	.45–.65						
Ash			.55–.71				
Fir	.65		.40	86	.094		
Elm			.56				
Hickory			.74–.80				
Mahogany			.56–.85				
Maple			.53–.68	86	.092		
Pine	.67		.43–.67	86	.065–.085		
Spruce			.45				
Walnut			.59				
Wool			1.32	86	.022		
Zinc (cast)			7.1				.10
Zinc (galvanized)						75	.22–.28

Note: The emissivity of a surface varies with its temperature, its degree of roughness, and, if a metal, its degree of oxidation. Therefore it must be remembered, emissivity for surfaces whose degree of roughness and oxidation is not specified is only an approximate value, and a wide variation is possible.

Table 4. Insulating Materials

Conductivity, *k*, is in Btu per hr sq ft deg F per inch of thickness. Conductance, *c*, is in Btu per hr sq ft deg F for the thickness specified.

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance	
				k	or *c*
BLANKET & BATT INSULATIONS	Asphalt-glass wool pad	17.7	90	.287	
		17.7	60	.273	
		17.7	30	.256	
		17.7	0	.241	
		17.7	−30	.226	
		17.7	−60	.211	
	Chemically treated wood fibers held between layers of strong paper	3.62	70	.25	
	Aerated wood fiber mat	1.7	90	.272	
		1.7	60	.252	
		1.7	30	.236	
		1.7	0	.219	
		1.7	−30	.203	
		1.7	−60	.187	
	Eelgrass between strong paper	4.60	90	.26	
		3.40	90	.25	
	Flax fibers between strong paper	4.90	90	.28	
	Chemically treated mohair between Kraft paper & asbestos paper	5.76	71	.26	
	Chemically treated hog hair between Kraft paper & asbestos paper	7.70	71	.28	
	Hair felt between layers of paper	11.00	75	.25	
	Kapok between burlap or paper	1.00	90	.24	
	Stitched & creod expanding fibrous blanket	1.50	70	.27	
	Paper & asbestos fiber with emulsified asphalt & binder	4.2	94	.28	
	Cotton insulating batt	.875	72	.24	
	Felted cattle hair	13.00	90	.26	
		11.00	90	.26	
	Hair felt ¾ in. thick	10.9	90		.265
		10.9	60		.250
		10.9	30		.236
		10.9	0		.223
		10.9	−30		.210
		10.9	−60		.199
	Felted hair & asbestos	7.8	90	.28	
	Ground paper between two layers, each ⅜ in. thick made up of two layers of Kraft paper (sample ¾ in. thick)	12.1	75		.40
	Granulated mineral wool	6	85	.26	
	Mineral wool felt	5.9	58.6	.224	
		5.9	106.0	.252	
		8.5	59.3	.216	
		8.5	107.2	.238	
	Kold board	16.2	32.0	.283	
		16.2	89.9	.318	
	Wood fiber blanket	3.5	90	.266	
		3.5	60	.250	
		3.5	30	.236	
		3.5	0	.223	
		3.5	−30	.211	
		3.5	−60	.198	
LOOSE FILL TYPE	Made from ceiba fibers	1.90	75	.23	
		1.60	75	.24	
	Cotton fibers				
	Short staple linters fireproofed	6.25	90	.25	

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance
				k or *c*
LOOSE FILL TYPE		4.50	90	.24
	Cotton	2.65	90	.246
		2.65	60	.253
		2.65	30	.221
		2.65	0	.208
		2.65	−30	.195
		2.65	−60	.183
	Cotton fibers Short staple linters fireproofed	2.45	90	.24
		1.60	90	.26
	Cotton	1.53	90	.247
		1.53	60	.230
		1.53	30	.216
		1.53	0	.202
		1.53	−30	.188
		1.53	−60	.176
		0.95	90	.266
		0.95	60	.245
		0.95	30	.227
		0.95	0	.208
		0.95	−30	.191
		0.95	−60	.173
		.65	90	.30
	Excelsior	1.84	90	.343
		1.84	60	.313
		1.84	30	.286
		1.84	0	.261
		1.84	−30	.237
		1.84	−60	.213
	Fibrous material made from dolomite & silica	1.5	75	.27
	Fibrous material made from slag	9.40	103	.27
	Redwood bark	3.0	90	.31
		4.0	90	.282
		4.0	60	.263
		4.0	30	.247
		4.0	0	.229
		4.0	−30	.213
		4.0	−60	.198
	Redwood bark	5.0	75	.26
	Glass wool fiber .0003 to .0060 in. in diameter	1.5	75	.27
	Glass wool	1.65	90	.303
		1.65	60	.272
		1.65	30	.247
		1.65	0	.225
		1.65	−30	.203
		1.65	−60	.183
	Granular insulation made from combined silicate of lime & alumina	4.2	72	.24
	Expanded vermiculite			.48
	Expanded vermiculite (particle size—3×14)	6.2		.32
	Regranulated cork $\frac{3}{16}$-in. particles	8.10	90	.31
	Hand applied granular mineral wool 2 in. to 6 in. thick, horizontal position, no covering	6.05 to 7.13		.30 to .33
	Mineral wool	3.5	90	0.287
		3.5	60	0.271
		3.5	30	0.238
		3.5	0	0.216

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance	
				k	or c
LOOSE FILL TYPE		3.5	−30	0.198	
		3.5	−60	0.186	
	4-in. machine blown granular mineral wool, horizontal position, no covering	5.74		.30	
	Rockwool	10.0	90	.27	
	Nodulated mineral wool 1½ in. thick	10.0	90		.289
		10.0	60		.272
		10.0	30		.256
		10.0	0		.242
		10.0	−30		.227
		10.0	−60		.213
	Silica aerogel	7.6	90	.169	
		7.6	60	.160	
		7.6	30	.152	
		7.6	0	.145	
		7.6	−30	.138	
		7.6	−60	.133	
	Expanded vermiculite, fill	8.2	90	.476	
		8.2	60	.457	
		8.2	30	.438	
		8.2	0	.420	
		8.2	−30	.401	
		8.2	−60	.383	
INSULATING BOARD	Made from sugar cane fiber	13.5	70	.33	
	Made from corn stalks	15.0	71	.33	
	Made from exploded wood fiber	17.9	78	.32	
	Made from hard wood fiber	15.2	70	.32	
	Made from wood fiber	15.9	72	.33	
		15.00	70	.33	
			52	.33	
		8.5	72	.29	
		15.20		.33	
		16.90	90	.34	
	Made from licorice root	16.1	81	.34	
	½-in. insulating board without special finish	16.5	90	.33	
		to		to	
	Eleven samples	21.8		.40	
	1-in. insulating board	15.2		.34	
	Corkboard (average) 1.12 in. thick	7.07	−14	.233	
	Corkboard (average) 1.0 in. thick	7.07	0	.27	
		7.07	−108	.201	
	Corkboard (average) 1.04 in. thick	6.70	−21	.242	
		6.70	−115	.204	
	Glass fiber board 1.00 in. thick	11.0	−16	.213	
		11.0	−111	.168	
	Cellular glass block	10.3	90	0.422	
		10.3	60	0.404	
		10.3	30	0.387	
		10.3	0	0.371	
		10.3	−30	0.353	
		10.3	−60	0.337	
	Expanded glass block 1.07 in. thick	10.6	−22	.409	
		10.6	−110	.381	
	Vegetable fiber board 1.02 in. thick	14.4	−24	.284	
		14.4	−110	.248	
	Expanded rubber board 1.00 in. thick	4.9	−22	.206	
		4.9	−100	.173	

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance	
				k	or *c*
INSULATING BOARD	Shredded redwood bark 1.00 in. thick	4.0	−25	.208	
		4.0	−119	.172	
	Mineral wool, board	15.7	90	.322	
		15.7	60	.308	
		15.7	30	.294	
		15.7	0	.281	
		15.7	−30	.268	
		15.7	−60	.254	
	Mineral board .99 in. thick wool	14.3	−21	.208	
		14.3	−112	.189	
	Vermiculite board	18.9	90	.927	
		18.9	60	.908	
		18.9	30	.889	
		18.9	0	.870	
		18.9	−30	.852	
		18.9	−60	.833	
	Wood fiber board ¾ in. thick, 24.1% moisture	15.7	90		.424
		15.7	60		.408
		15.7	30		.392
		15.7	0		.376
		15.7	−30		.358
		15.7	−60		.343
	Wood fiber board ¾ in. thick, 3.1% moisture	15.7	90		.356
		15.7	60		.343
		15.7	30		.331
		15.7	0		.319
		15.7	−30		.306
		15.7	−60		.294
	Wood fiber board ¾ in. thick, 0.0% moisture	15.7	90		.348
		15.7	60		.335
		15.7	30		.323
		15.7	0		.310
		15.7	−30		.298
		15.7	−60		.285
SLAB INSULATIONS	Corkboard (no added binder)	14.0	90	.34	
		10.6	90	.30	
		7.0	90	.27	
		5.4	90	.25	
	Corkboard	8.7	—	.29	
	Corkboard (oven dry)	12.2	90	.309	
		12.2	60	.300	
		12.2	30	.292	
		12.2	0	.282	
		12.2	−30	.273	
		12.2	−60	.264	
		8.0	90	.276	
		8.0	60	.267	
		8.0	30	.258	
		8.0	0	.249	
		8.0	−30	.240	
		8.0	−60	.231	
		6.5	90	.277	
		6.5	60	.266	
		6.5	30	.255	
		6.5	0	.244	
		6.5	−30	.233	
		6.5	−60	.222	

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance	
				k or	*c*
SLAB INSULATIONS	Corkboard asphaltic binder	14.5	90	.32	
	Chemically treated hog hair with film of asphalt	10.0	75	.28	
	Sugar cane fiber insulating blocks encased in asphalt membrane	13.8	70	.30	
	Made from 85% magnesia and 15% asbestos	19.3	86	.51	
	Made from shredded wood & cement	24.2	72	.46	
		29.8		.77	
BUILDING BOARDS (Non-insulating)	Compressed cement & asbestos sheets	123	86	2.70	
	Corrugated asbestos board	20.4	110	0.48	
	Lignin-impregnated cellulose fiber board 1¼ in. thick	85.5	90		1.84
		85.5	60		1.81
		85.5	30		1.77
		85.5	0		1.74
		85.5	−30		1.71
		85.5	−60		1.67
	Pressed asbestos mill board	60.5	86	.84	
	Gypsum board—gypsum between paper	62.8	70	1.41	
	⅜-in. gypsum board				3.73
	½-in. gypsum board				2.82
	½-in. gypsum board	53.5	90		2.60
CLAY TILE, HOLLOW	2-in. tile, ½-in. plaster both sides	120.0	110		1.00
	4-in. tile, ½-in. plaster both sides	127.0	100		.60
	6-in. tile, ½-in. plaster both sides	124.3	105		.47
	8-in. tile, average of 8 types				.52
	12-in. clay tile wall				
	8×5×12				
	4×5×12				.26
CONCRETE	Sand & gravel aggregate, various ages and mixes			11.35 to 16.36	
	Sand & gravel aggregate	142	75	12.6	
	Limestone aggregate	132	75	10.8	
	Cinder aggregate	97	75	4.9	
	Steam treated limestone slag aggregate	74.6	75	2.27	
	Pumice aggregate	65.0	75	2.42	
	Expanded burned clay aggregate	59.9	75	2.28	
	Burned clay aggregate	67.1	75	2.86	
	Blast furnace slag aggregate	76.0	70	1.6	
	Expanded vermiculite aggregate	20	90	.68	
		26.7	90	.76	
		35	90	.86	
		50	90	1.10	
	Concrete plank	76	75	2.5	
	Cellular concrete	40.0	75	1.06	
		50	75	1.44	
		60	75	1.80	
		70	75	2.18	
CONCRETE BLOCKS 8-in. concrete blocks 8×8×16 3 oval	8-in. three oval core sand & gravel aggregate	126.4	40		.90
	8-in. three oval core crushed limestone	134.3	40		.86
	8-in. three oval core burned clay aggregate	67.7	40		.50
	8-in. three oval core cinder aggregate	86.2	40		.58
	8-in. three oval core expanded blast furnace aggregate		40		.49

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k	Conductivity or conductance c
12-in. concrete blocks 8×12×16 3 oval	12-in. three oval core sand & gravel aggregate	124.9	40		.78
	12-in. three oval core cinder aggregate	86.2	40		.53
	12-in. three oval core burned clay	76.7	40		.47
FRAME CONSTRUCTION COMBINATIONS	1-in. fir sheathing and building paper		30		.86
	1-in. fir sheathing building paper and yellow pine siding		20		.50
	1-in. fir sheathing building paper and stucco		20		.82
	Pine lap siding and building paper, siding 4 in. wide		16		.85
	Yellow pine lap siding				1.28
GYPSUM	3-in. solid gypsum partitioned tile			2.41	
	3-in. three cell gypsum partitioned tile				.74
	4-in. three cell gypsum partitioned tile				.60
	87½%, gypsum, 12½% wood chips	51.2	74	1.66	
	Gypsum plaster			3.30	
MASONRY MATERIALS BRICK	Damp or wet			5.0	
	Common yellow clay brick			4.8	
	One tier yellow common clay brick, one-tier face brick, approx. 8 in. thick				.77
PLASTERING MATERIALS	Gypsum plaster ⅜ in. thick		73		8.8
	Cement plaster			12.00	
	Wood lath & plaster, total thickness ¾ in.		70		2.50
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	.85	
	Insulating plaster 0.9 in. thick applied to ⅜-in. gypsum board	54.0	75		1.07
ROOFING	Asbestos shingles	65.0	75		6.0
	Asphalt composition or prepared	70	75		6.5
	Asphalt shingles	70	75		6.5
	Built up roofing, bitumen or felt, gravel or slag surfaced			1.33	
	Slate			10.0	
	Wood shingles			1.28	
WOODS	Balsa	20.0	90	.58	
		8.8	90	.38	
		7.3	90	.33	
	California redwood, 0% moisture	28.0	75	.70	
	Cypress	28.7	86	.67	
	Douglas fir, 0% moisture	34	75	.67	
	Eastern hemlock, 0% moisture	30	75	.76	
	Fir plywood, ¾ in. thick	33.2	90		.813
		33.2	60		.791
		33.2	30		.768
		33.2	0		.745
		33.2	−30		.723
		33.2	−60		.702
	Fir plywood, four ¼ in. thick, 17.3% moisture	31.9	90		.896
		31.9	60		.873
		31.9	30		.847
		31.9	0		.823

Table 4. Insulating Materials (Concluded)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance	
				k or	*c*
WOODS		31.9	−30		.800
		31.9	−60		.776
	Fir plywood, four ¼ in. thick, 2.5% moisture	31.9	90		.735
		31.9	60		.714
		31.9	30		.693
		31.9	0		.672
		31.9	−30		.650
		31.9	−60		.629
	Fir plywood, four ¼ in. thick, 0% moisture	31.9	90		.707
		31.9	60		.683
		31.9	30		.659
		31.9	0		.636
		31.9	−30		.612
		31.9	−60		.589
	Long leaf yellow pine, 0% moisture	40	75	.86	
	Mahogany	34.3	86	.90	
	Hard maple, 0% moisture	46.0	75	1.05	
	Maple	44.3	86	1.10	
	Maple crossgrain	40.0	75	1.20	
	Norway pine, 0% moisture	32.0	75	.74	
	Pine 1¼ in. thick	24.1	90		.707
		24.1	60		.689
		14.1	30		.670
		24.1	0		.651
		24.1	−30		.633
		24.1	−60		.614
	Red cypress, 0% moisture	32.0	75	.77	
	Short leaf yellow pine, 0% moisture	36.0	75	.91	
	Soft elm, 0% moisture	34.0	75	.88	
	Soft maple, 0% moisture	42.0	75	.95	
	Sugar pine, 0% moisture	28.0	75	.64	
	Virginia pine	34.3	86	.96	
	West coast hemlock, 0% moisture	30.0	75	.79	
	White pine	31.2	86	.78	
	Yellow pine			1.00	
	Sawdust various	12.0	90	.41	
	Shavings from planer	8.8	90	.41	
	Shavings from maple, beech & birch	13.2	90	.36	
	Air space-bounded by ordinary materials 1 in. thick		90	1.18	
			60	1.07	
			30	.98	
			0	.89	
			−30	.82	
			−60	.74	
	Air space—bounded warm side by aluminum paint 1 in. thick		90	.64	
			60	.62	
			30	.59	
			0	.57	
			−30	.55	
			−60	.53	
	Air space—bounded warm side by aluminum foil 1 in. thick		90	.47	
			60	.46	
			30	.46	
			0	.45	
			−30	.45	
			−60	.45	

Table 5. Food Products Data

Name	Average freezing point F	% water	Sp ht, Btu/lb F		Latent heat of fusion Btu/lb	Heat of respiration Btu/24 hr ton F	
			Above freezing	Below freezing			
VEGETABLES							
Artichokes	29.1	83.7	.87	.45	120	40	10,140
Asparagus	29.8	93	.94	.48	134	40	11,500
Beans, string	29.7	88.9	.91	.47	128	32 40	4,740 6,740
Beans, Lima	30.1	66.5	.73	.40	94		
Beans, dried		12.5	.30	.24	18		
Beets	31.1	90	.86	.47	129	32 40	1,166 1,820
Broccoli	29.2	89.9	.92	.47	130		
Brussels sprouts	31	84.9	.88	.46	122		
Cabbage	31.2	92.4	.94	.47	132		
Carrots	2916	88.2	.86	.45	126	32 40	2,130 3,470
Cauliflower	30.1	91.7	.93	.47	132		
Celery	29.7	93.7	.95	.48	135	32 40	2,820 4,540
Corn (green)	28.9	75.5	.80	.43	108	32 40	5,890 8,190
Corn (dried)		10.5	.28	.23	15		
Cucumbers	30.5	96.1	.97	.49	137		
Eggplant	30.4	92.7	.94	.47	132		
Endive (escarole)	30.9	93.3	.94	.48	132		
Lettuce	31.2	94.8	.96	.48	136	32 40	638 7,392
Mushrooms	30.2	91.1	.93	.47	130	32 50	6,160 22,000
Onions	30.1	87.5	.91	.46	124	32 50	880 1,870
Parsnips	28.9	78.6	.84	.46	112		
Peas (green)	30	74.3	.79	.42	106		
Peas (dried)		9.5	.28	.22	14		
Peppers (sweet)	30.1	92.4	.94	.47		40	1,518
Potatoes (white)	28.9	77.8	.82	.43	111	32 40	660 1,430
Potatoes (sweet)	28.5	68.5	.75	.40	97	40	1,710
Pumpkin	30.1	90.5	.92	.47	130		
Radishes		93.6	.95	.48	134		
Rhubarb	28.4	94.9	.96	.48	134		
Sauerkraut	26	89	.92	.47	129		
Spinach	30.3	92.7	.94	.48	132		
Squash	30.1	90.5	.92	.47	130		
Tomatoes (green)	30.4	94.7	.95	.48	134	60	6,230
Tomatoes (ripening)	30.4	94.1	.95	.48	134	40	1,260
Turnips	30.5	90.9	.93	.40	137	32 40	66 572
Vegetables (mixed)	30	90	.90	.45	130		
MEATS & FISH							
Bacon		20	.50	.30	29	28 40	2,400 5,000
Beef (dried)		5-15	.22-.34	.19-.26	7-22	36 45	186 280
Beef (fresh-lean)	29	68	.77	.40	100	30 35	3,400 10,000
Beef (fresh-fat)	28		.60	.35	79		
Brined Meats			.75			31 40	1,540 1,940

Table 5. Food Products Data (Continued)

Name	Average freezing point F	% water	Sp ht, Btu/lb F		Latent heat of fusion, Btu/lb	Heat of respiration, Btu/24 hr ton F	
			Above freezing	Below freezing			
Cut meats	29	65	.72	.40	95	34	11,200
Fish (frozen)	28	70	.76	.41	101	5	200
						15	2,040
Fish (iced)		70	.76	.41	101	30	840
						34	1,140
Fish (dried)			.56	.34	65		
Hams & loins	27	60	.68	.38	86.5	28	3,500
		58				34	6,800
Lamb	29		.67	.30	83.5	34	6,720
		65.5				28	2,500
Livers	29		.72	.40	93.3	28	7,200
		80.4				20	1,506
Oysters (shell)	27		.83	.44	116	35	8,300
		87				32	1,020
Oysters (tub)	27		.90	.46	125	35	4,500
		60				32	440
Pork (fresh)	28	57	.68	.38	86.5	34	6,800
Pork (smoked)		74	.60	.32		55	2,500
Poultry (fresh)	27	74	.79	.37	106	28	6,880
Poultry (frozen)	27		.79	.37	106	15	2,480
							480
Sausage (casings)		65.5	.60				
Sausage (drying)	26	60	.89	.56	93		
Sausage (franks)	29	65	.86	.56	86	45	8,600
Sausage (fresh)	26	60	.89	.56	93	35	8,600
Sausage (smoked)	25	63	.86	.56	86	45	6,440
Veal	29		.71	.39	91	34	7,010
						28	2,500
MISCELLANEOUS		.92					
Beer (wood kegs)	28	.92	1.0				
Beer (steel kegs)	28	15	1.0				
Butter	30–0		.64	.34	15	45	3,960
						35	3,640
Caviar (tub)	20	55				40	3,820
Cheese (American)	17	60	.64	.36	79	40	4,680
Cheese (Camembert)	18	60	.70	.40	86	40	4,920
Cheese (Limburger)	19	55	.70	.40	86	40	4,920
Cheese (Roquefort)	3	55	.65	.32	79	45	4,000
Cheese (Swiss)	15	.5	.64	.36	79	40	4,660
Chocolate (coating)	95–85	55	.30	.55	40		
Cream (40%)	28	73	.85	.40	90		
Eggs (crated)	27		.76	.40	100	40	7,010
Eggs (frozen)	27			.41	100		
Flowers (cut)	32						480/Floor
Furs—Woolens		18		.40			
Honey			.35	.26	26	40	1,420
Hops						35	1,500
Ice cream	27–0	10	.78	.45	96		
Lard			.52			45	3,960
Malt		5				50	1,500
Maple sugar		36	.24	.21	7	45	1,420
Maple syrup		87.5	.49	.31	52	45	1,420
Milk	31	3–10	.93	.49	124		
Nuts (dried)			.21–.29	.19–.24	4.3–14	35	1,000
Oleomargarine			.48			45	3,960
Tobacco & cigars	25						
Candy			.93				
Flour		13.5	.38	.28			

Table 5. Food Products Data (Concluded)

Name	Average freezing point F	% water	Sp ht, Btu/lb F		Latent heat of fusion Btu/lb	Heat of respiration, Btu/24 hr ton F	
			Above freezing	Below freezing			
FRUITS							
Apples	28.4	84.1	.86	.45	121	32	830
						40	1,435
Apricots	28.1	85.4	.88	.46	122		
Avocadoes	27.2	94	.91	.49	136	67	8,360
Bananas	28	74.8	.80	.42	108		
Blackberries	28.9	85.3	.88	.46	122		
Cantaloupes	29	92.7	.94	.48	132	40	3,470
						60	8,080
Cherries	26	83	.87	.45	120		
Cranberries	27.3	87.4	.90	.46	124		
Currants	30.2	84.7	.88	.45	120		
Dates (dry—cured)	−4.1	20	.36	.26	29		
Dates (dry—non-cured)	−4.1	20	.36	.26	29		
Dates (fresh)	27.1	78	.82	.43	112		
Figs (fresh)	27.1	78	.82	.43	112		
Figs (dried)		24	.39	.27	34		
Gooseberries	28.9	88.3	.90	.46	126		
Grapefruit	28.4	88.8	.91	.46	126	32	460
						40	1,070
Grapes	26.3	81.7	.86	.44	116	35	830
Honey Dew Melon	20	92.6	.94	.48	132		
Lemons	28.1	89.3	.92	.46	127	40	810
						60	2,970
Limes	29	86	.89	.46	122	40	810
						60	2,970
Mangoes	32	93	.90	.46	134		
Oranges	28	87.2	.90	.46	124	32	795
						40	1,400
Peaches	29.4	86.9	.90	.46	124	32	1,110
						40	1,735
Pears	28.5	83.5	.86	.45	118	32	770
Persimmons	28.3	78.2	.84	.43	112		
Pineapples	29.4	85.3	.88	.45	122		
Plums	28	85.7	.88	.45	123		
Pomegranates	28	77	.87	.48	112		
Prunes (fresh)	28	85.7	.88	.45	123		
Quinces	28.1	85.3	.88	.45	122		
Raspberries	30.1	82	.85	.45	122	36	5,500
						60	16,500
Strawberries	29.9	90	.92	.47	129	32	3,265
						40	5,865
Watermelons	29.2	92.1	.97	.48	132		

Table 6. Properties of Ice*

Temp F	Sp vol cu ft per lb	Density		Enthalpy of fusion Btu per lb
		lb per cu ft	lb per gallon	
32	0.01747	57.24	7.65	143.35
20	0.01747	57.24	7.65	
0	0.01742	57.40	7.67	
−20	0.01739	57.50	7.69	

Sp ht of ice 32 F to 20 F range, 0.496 Btu per lb F
20 F to 0 F range, 0.481 Btu per lb F
0 F to −40 F range, 0.452 Btu per lb F
−40 F to −80 F range, 0.414 Btu per lb F

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* Prepared by B. H. Jennings.

Table 7. Permeability of Various Materials to Water Vapor

(Units—Grains per sq ft per day per mm of Hg)

Item No.	Material	Thickness, in.	Investigator and permeability			
			Babbitt (dry method)	Miscellaneous	J-M Research Laboratories Humid method	J-M Research Laboratories Dry method
	PAPER AND FELTS					
1	Kraft paper, 1 sheet	0.00394	74.3			
2	Kraft paper, 2 sheets in series		47.2			
3	Kraft paper, 3 sheets in series		35.4			
4	Kraft paper, 4 sheets in series		28.1			
5	Kraft paper, 5 sheets in series		26.4			
6	Uncoated kraft (about 30-lb)				130	
7	Rosin sized sheathing paper (30–40 lb)				91	
8	Asphalt saturated rag felts (15–20 lb)		0.4 –30		4.0 –30	
9	Asphalt saturated and coated kraft papers		0.45	0.17– 2.8T	0.0 – 3.0	
10	Asphalt saturated asbestos felt	0.03–0.04			1.0 – 5.0	
11	Asphalt saturated and coated asbestos felt, 20-lb	0.03–0.04	0.14		1.0 – 2.0	
12	Tarred slaters felt	0.03–0.04	4–10	5.2 –25.0T	40–45	
13	35-lb specification cold application asbestos felt	0.032			0.0	
14	No. 50 asbestos waterproofing felt	0.051			0.9	
15	asph sat. rfg felt (30-lb) 1 sheet	0.062			5.1	1.7
6	asph sat. rfg felt (30-lb) 2 sheets in series				2.5	1.1
17	asph sat. rfg felt (30-lb) 3 sheets in series				1.4	0.87
18	Duplex papers		0.3 – 1.5	0.57– 2.5T	0.2 – 6.0	
19	Waxed papers		0.1 – 4.5			
20	Heavy roofing sheets		0.03– 1.4	0.13– 0.17T	0.03– 1.0	
	MISCELLANEOUS MATERIALS					
21	Wood	0.5	1.74– 2.02			
22	Wood	1.0	1.08– 1.25			
23	Insulating board	0.5	36.4			
24	Insulating board	1.0	20.6			
25	Insulating board, Plain	25/32		26.0–35.0T	30.0	
26	Insulating board, Sealed or Coated	25/32		3.1 – 4.3T	0.20–10.0	
27	Rock Cork	0.523			5.6	2.8
28	Rock Cork	0.984			4.2	2.3
29	Rock Cork	2.001			2.7	1.9
30	Pure corkboard (insulating)	1.0		6.2T	9.0	
31	Plaster on metal lath	0.5		15.2R		
32	Plastei on metal lath	0.75		12.8R		
33	Plaster on wood lath	0.5		6.4R, 3.6–3.9T		
34	Plaster ½-in. on ⅜-in. gypsum board			10.7R		
35	Plaster ½-in. on ½-in. wood fiber board			15.5R		
36	Stucco	1.0			2.0	
37	Marinite	0.625			7.5	
38	Transite	0.25			6.0	
39	Flexboard	0.125			15.5	5.5
40	1½-in. Concrete (1:2:4 Mix)				2.7	
41	Concrete	1.5		15.7B		
42	Brick wall section—with mortar	4.0		7.5B		
43	Tile wall section—with mortar	4.0		1.2B		
	COMBINATIONS					
44	⅛-in. Zeroseal on ½-in. J-M Rock Cork				0.16– 0.21	
45	⅛-in. Zerokote on ½-in. J-M Rock Cork				2.4	
46	½-in. Portland cement mortar (1:3 Mix) on ½-in. J-M Rock Cork				17.6	
47	½-in. Rock Cork dipped in asphalt, then applied to 15-lb roofing felt nailed to wood sheathing				0.51– 0.75	
48	½-in. Rock Cork dipped in hot asphalt, then applied to 1½-in. concrete—concrete primer				0.24	
49	½-in. Rock Cork dipped in hot asphalt, then applied to Vaporseal paper nailed to wood sheathing				0.00	

Note: Under Miscellaneous: B denotes Barre—temperature gradient method; R denotes Rowley—temperature gradient method; T denotes Teesdale—humid method.

13. MATHEMATICAL TABLES

Table 1. Natural Trigonometric Functions

Degrees	Sines							Cosines
	0′	10′	20′	30′	40′	50′	60′	
0	0.00000	0.00291	0.00582	0.00873	0.01164	0.01454	0.01745	89
1	0.01745	0.02036	0.02327	0.02618	0.02908	0.03199	0.03490	88
2	0.03490	0.03781	0.04071	0.04362	0.04653	0.04943	0.05234	87
3	0.05234	0.05524	0.05814	0.06105	0.06395	0.06685	0.06976	86
4	0.06976	0.07266	0.07556	0.07846	0.08136	0.08426	0.08716	85
5	0.08716	0.09005	0.09295	0.09585	0.09874	0.10164	0.10453	84
6	0.10453	0.10742	0.11031	0.11320	0.11609	0.11898	0.12187	83
7	0.12187	0.12476	0.12764	0.13053	0.13341	0.13629	0.13917	82
8	0.13917	0.14205	0.14493	0.14781	0.15069	0.15356	0.15643	81
9	0.15643	0.15931	0.16218	0.16505	0.16792	0.17078	0.17365	80
10	0.17365	0.17651	0.17937	0.18224	0.18509	0.18795	0.19081	79
11	0.19081	0.19366	0.19652	0.19937	0.20222	0.20507	0.20791	78
12	0.20791	0.21076	0.21360	0.21644	0.21928	0.22212	.22495	77
13	0.22495	0.22778	0.23062	0.23345	0.23627	0.23910	.24192	76
14	0.24192	0.24474	0.24756	0.25038	0.25320	0.25601	0.25882	75
15	0.25882	0.26163	0.26443	0.26724	0.27004	0.27284	0.27564	74
16	0.27564	0.27843	0.28123	0.28402	0.28680	0.28959	0.29237	73
17	0.29237	0.29515	0.29793	0.30071	0.30348	0.30625	0.30902	72
18	0.30902	0.31178	0.31454	0.31730	0.32006	0.32282	0.32557	71
19	0.32557	0.32832	0.33106	0.33381	0.33655	0.33929	0.34202	70
20	0.34202	0.34475	0.34748	0.35021	0.35293	0.35565	0.35837	69
21	0.35837	0.36108	0.36379	0.36650	0.36921	0.37191	0.37461	68
22	0.37461	0.37730	0.37999	0.38268	0.38537	0.38805	0.39073	67
23	0.39073	0.39341	0.39608	0.39875	0.40142	0.40408	0.40674	66
24	0.40674	0.40939	0.41204	0.41469	0.41734	0.41998	0.42262	65
25	0.42262	0.42525	0.42788	0.43051	0.43313	0.43575	0.43837	64
26	0.43837	0.44098	0.44359	0.44620	0.44880	0.45140	0.45399	63
27	0.45399	0.45658	0.45917	0.46175	0.46433	0.46690	0.46947	62
28	0.46947	0.47204	0.47460	0.47716	0.47971	0.48226	0.48481	61
29	0.48481	0.48735	0.48989	0.49242	0.49495	0.49748	0.50000	60
30	0.50000	0.50252	0.50503	0.50754	0.51004	0.51254	0.51504	59
31	0.51504	0.51753	0.52002	0.52250	0.52498	0.52745	0.52992	58
32	0.52992	0.53238	0.53484	0.53730	0.53975	0.54220	0.54464	57
33	0.54464	0.54708	0.54951	0.55194	0.55436	0.55678	0.55919	56
34	0.55919	0.56160	0.56401	0.56641	0.56880	0.57119	0.57358	55
35	0.57358	0.57596	0.57833	0.58070	0.58307	0.58543	0.58779	54
36	0.58779	0.59014	0.59248	0.59482	0.59716	0.59949	0.60182	53
37	0.60182	0.60414	0.60645	0.60876	0.61107	0.61337	0.61566	52
38	0.61566	0.61795	0.62024	0.62251	0.62479	0.62706	0.62932	51
39	0.62932	0.63158	0.63383	0.63608	0.63832	0.64056	0.64279	50
40	0.64279	0.64501	0.64723	0.64945	0.65166	0.65386	0.65606	49
41	0.65606	0.65825	0.66044	0.66262	0.66480	0.66697	0.66913	48
42	0.66913	0.67129	0.67344	0.67559	0.67773	0.67987	0.68200	47
43	0.68200	0.68412	0.68624	0.68835	0.69046	0.69256	0.69466	46
44	0.69466	0.69675	0.69883	0.70091	0.70298	0.70505	0.70711	45
Sines	60′	50′	40′	30′	20′	10′	0′	Degrees
	Cosines							

Table 1. Natural Trigonometric Functions (Continued)

Degrees	Cosines							Sines
	0′	10′	20′	30′	40′	50′	60′	
0	1.00000	1.00000	0.99998	0.99996	0.99993	0.99989	0.99985	89
1	0.99985	0.99979	0.99973	0.99966	0.99958	0.99949	0.99939	88
2	0.99939	0.99929	0.99917	0.99905	0.99892	0.99878	0.99863	87
3	0.99863	0.99847	0.99831	0.99813	0.99795	0.99776	0.99756	86
4	0.99756	0.99736	0.99714	0.99692	0.99668	0.99644	0.99619	85
5	0.99619	0.99594	0.99567	0.99540	0.99511	0.99482	0.99452	84
6	0.99452	0.99421	0.99390	0.99357	0.99324	0.99290	0.99255	83
7	0.99255	0.99219	0.99182	0.99144	0.99106	0.99067	0.99027	82
8	0.99027	0.98986	0.98944	0.98902	0.98858	0.98814	0.98769	81
9	0.98769	0.98723	0.98676	0.98629	0.98580	0.98531	0.98481	80
10	0.98481	0.98430	0.98378	0.98325	0.98272	0.98218	0.98163	79
11	0.98163	0.98107	0.98050	0.97992	0.97934	0.97875	0.97815	78
12	0.97815	0.97754	0.97692	0.97630	0.97566	0.97502	0.97437	77
13	0.97437	0.97371	0.97304	0.97237	0.97169	0.97100	0.97030	76
14	0.97030	0.96959	0.96887	0.96815	0.96742	0.96667	0.96593	75
15	0.96593	0.96517	0.96440	0.96363	0.96285	0.96206	0.96126	74
16	0.96126	0.96048	0.95964	0.95882	0.95799	0.95715	0.95630	73
17	0.95630	0.95545	0.95459	0.95372	0.95284	0.95195	0.95106	72
18	0.95106	0.95015	0.94924	0.94832	0.94740	0.94646	0.94552	71
19	0.94552	0.94457	0.94361	0.94264	0.94167	0.94068	0.93969	70
20	0.93969	0.93869	0.93769	0.93667	0.93565	0.93462	0.93358	69
21	0.93358	0.93253	0.93148	0.93042	0.92935	0.92827	0.92718	68
22	0.92718	0.92609	0.92499	0.92388	0.92276	0.92164	0.92050	67
23	0.92050	0.91936	0.91822	0.91706	0.91590	0.91472	0.91355	66
24	0.91355	0.91236	0.91116	0.90996	0.90875	0.90753	0.90631	65
25	0.90631	0.90507	0.90383	0.90259	0.90133	0.90007	0.89879	64
26	0.89879	0.89752	0.89623	0.89493	0.89363	0.89232	0.89101	63
27	0.89101	0.88968	0.88835	0.88701	0.88566	0.88431	0.88295	62
28	0.88295	0.88158	0.88020	0.87882	0.87743	0.87603	0.87462	61
29	0.87462	0.87321	0.87178	0.87036	0.86892	0.86748	0.86603	60
30	0.86603	0.86457	0.86310	0.86163	0.86015	0.85866	0.85717	59
31	0.85717	0.85567	0.85416	0.85264	0.85112	0.84959	0.84805	58
32	0.84805	0.84650	0.84495	0.84339	0.84182	0.84025	0.83867	57
33	0.83867	0.83708	0.83549	0.83389	0.83228	0.83066	0.82904	56
34	0.82904	0.82741	0.82577	0.82413	0.82248	0.82082	0.81915	55
35	0.81915	0.81748	0.81580	0.81412	0.81242	0.81072	0.80902	54
36	0.80902	0.80730	0.80558	0.80386	0.80212	0.80038	0.79864	53
37	0.79864	0.79688	0.79512	0.79335	0.79158	0.78980	0.78801	52
38	0.78801	0.78622	0.78442	0.78261	0.78079	0.77897	0.77715	51
39	0.77715	0.77531	0.77347	0.77162	0.76977	0.76791	0.76604	50
40	0.76604	0.76417	0.76229	0.76041	0.75851	0.75661	0.75471	49
41	0.75471	0.75280	0.75088	0.74896	0.74703	0.74509	0.74314	48
42	0.74314	0.74120	0.73924	0.73728	0.73531	0.73333	0.73135	47
43	0.73135	0.72937	0.72737	0.72537	0.72337	0.72136	0.71934	46
44	0.71934	0.71732	0.71529	0.71325	0.71121	0.70916	0.70711	45
Cosines	60′	50′	40′	30′	20′	10′	0′	Degrees
	Sines							

Table 1. Natural Trigonometric Functions (Concluded)

Degrees	Tangents							Co-tangents
	0′	10′	20′	30′	40′	50′	60′	
0	0.00000	0.00291	0.00582	0.00873	0.01164	0.01455	0.01746	89
1	0.01746	0.02036	0.02328	0.02619	0.02910	0.03201	0.03492	88
2	0.03492	0.03783	0.04075	0.04366	0.04658	0.04949	0.05241	87
3	0.05241	0.05533	0.05824	0.06116	0.06408	0.06700	0.06993	86
4	0.06993	0.07285	0.07578	0.07870	0.08163	0.08456	0.08749	85
5	0.08749	0.09042	0.09335	0.09629	0.09923	0.10216	0.10510	84
6	0.10510	0.10805	0.11099	0.11394	0.11688	0.11983	0.12278	83
7	0.12278	0.12574	0.12869	0.13165	0.13461	0.13758	0.14054	82
8	0.14054	0.14351	0.14648	0.14945	0.15243	0.15540	0.15838	81
9	0.15838	0.16137	0.16435	0.16734	0.17033	0.17333	0.17633	80
10	0.17633	0.17933	0.18233	0.18534	0.18835	0.19136	0.19438	79
11	0.19438	0.19740	0.20042	0.20345	0.20648	0.20952	0.21256	78
12	0.21256	0.21560	0.21864	0.22169	0.22475	0.22781	0.23087	77
13	0.23087	0.23393	0.23700	0.24008	0.24316	0.24624	0.24933	76
14	0.24933	0.25242	0.25552	0.25862	0.26172	0.26483	0.26795	75
15	0.26795	0.27107	0.27419	0.27732	0.28046	0.28360	0.28675	74
16	0.28675	0.28990	0.29305	0.29621	0.29938	0.30255	0.30573	73
17	0.30573	0.30891	0.31210	0.31530	0.31850	0.32171	0.32492	72
18	0.32492	0.32814	0.33136	0.33460	0.33783	0.34108	0.34433	71
19	0.34433	0.34758	0.35085	0.35412	0.35740	0.36068	0.36397	70
20	0.36397	0.36727	0.37057	0.37388	0.37720	0.38053	0.38386	69
21	0.38386	0.38721	0.39055	0.39391	0.39727	0.40065	0.40403	68
22	0.40403	0.40741	0.41081	0.41421	0.41763	0.42105	0.42447	67
23	0.42447	0.42791	0.43136	0.43481	0.43828	0.44175	0.44523	66
24	0.44523	0.44872	0.45222	0.45573	0.45924	0.46277	0.46631	65
25	0.46631	0.46985	0.47341	0.47698	0.48055	0.48414	0.48773	64
26	0.48773	0.49134	0.49495	0.49858	0.50222	0.50587	0.50953	63
27	0.50953	0.51320	0.51688	0.52057	0.52427	0.52798	0.53171	62
28	0.53171	0.53545	0.53920	0.54296	0.54674	0.55051	0.55431	61
29	0.55431	0.55812	0.56194	0.56577	0.56962	0.57348	0.57735	60
30	0.57735	0.58124	0.58513	0.58905	0.59297	0.59691	0.60086	59
31	0.60086	0.60483	0.60881	0.61280	0.61681	0.62083	0.62487	58
32	0.62487	0.62892	0.63299	0.63707	0.64117	0.64528	0.64941	57
33	0.64941	0.65355	0.65771	0.66189	0.66608	0.67028	0.67451	56
34	0.67451	0.67875	0.68301	0.68728	0.69157	0.69588	0.70021	55
35	0.70021	0.70455	0.70891	0.71329	0.71769	0.72211	0.72654	54
36	0.72654	0.73100	0.73547	0.73996	0.74447	0.74900	0.75355	53
37	0.75355	0.75812	0.76272	0.76733	0.77196	0.77661	0.78129	52
38	0.78129	0.78598	0.79070	0.79544	0.80020	0.80498	0.80978	51
39	0.80978	0.81461	0.81946	0.82434	0.82923	0.83415	0.83910	50
40	0.83910	0.84407	0.84906	0.85408	0.85912	0.86419	0.86929	49
41	0.86929	0.87441	0.87955	0.88473	0.88992	0.89515	0.90040	48
42	0.90040	0.90569	0.91099	0.91633	0.92170	0.92709	0.93252	47
43	0.93252	0.93797	0.94345	0.94896	0.95451	0.96008	0.96569	46
44	0.96569	0.97133	0.97700	0.98270	0.98843	0.99420	1.00000	45
Tangents	60′	50′	40′	30′	20′	10′	0′	Degrees
	Cotangents							

Table 2. Common Logarithms, Base of 10

N	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	*4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	*3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	11
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7
N	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9

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* Interpolation in this section of the table is inaccurate.

Table 2. Common Logarithms (Concluded)

N	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4
N	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9

Table 3. Decimal of an Inch and of a Foot

Fractions of inch or foot		Inch equivalents to foot fractions	Fractions of inch or foot		Inch equivalents to foot fractions	Fractions of inch or foot		Inch equivalents to foot fractions	Fractions of inch or foot		Inch equivalents to foot fractions
	.0052	1/16		.2552	3 1/16		.5052	6 1/16		.7552	9 1/16
	.0104	1/8		.2604	3 1/8		.5104	6 1/8		.7604	9 1/8
1/64	.015625	3/16	17/64	.265625	3 3/16	33/64	.515625	6 3/16	49/64	.765625	9 3/16
	.0208	1/4		.2708	3 1/4		.5208	6 1/4		.7708	9 1/4
	.0260	5/16		.2760	3 5/16		.5260	6 5/16		.7760	9 5/16
1/32	.03125	3/8	9/32	.28125	3 3/8	17/32	.53125	6 3/8	25/32	.78125	9 3/8
	.0365	7/16		.2865	3 7/16		.5365	6 7/16		.7865	9 7/16
	.0417	1/2		.2917	3 1/2		.5417	6 1/2		.7917	9 1/2
3/64	.046875	9/16	19/64	.296875	3 9/16	35/64	.546875	6 9/16	51/64	.796875	9 9/16
	.0521	5/8		.3021	3 5/8		.5521	6 5/8		.8021	9 5/8
	.0573	11/16		.3073	3 11/16		.5573	6 11/16		.8073	9 11/16
1/16	.0625	3/4	5/16	.3125	3 3/4	9/16	.5625	6 3/4	13/16	.8125	9 3/4
	.0677	13/16		.3177	3 13/16		.5677	6 13/16		.8177	9 13/16
	.0729	7/8		.3229	3 7/8		.5729	6 7/8		.8229	9 7/8
5/64	.078125	15/16	21/64	.328125	3 15/16	37/64	.578125	6 15/16	53/64	.828125	9 15/16
	.0833	1		.3333	4		.5833	7		.8333	10
	.0885	1 1/16		.3385	4 1/16		.5885	7 1/16		.8385	10 1/16
3/32	.09375	1 1/8	11/32	.34375	4 1/8	19/32	.59375	7 1/8	27/32	.84375	10 1/8
	.0990	1 3/16		.3490	4 3/16		.5990	7 3/16		.8490	10 3/16
	.1042	1 1/4		.3542	4 1/4		.6042	7 1/4		.8542	10 1/4
7/64	.109375	1 5/16	23/64	.359375	4 5/16	39/64	.609375	7 5/16	55/64	.859375	10 5/16
	.1146	1 3/8		.3646	4 3/8		.6146	7 3/8		.8646	10 3/8
	.1198	1 7/16		.3698	4 7/16		.6198	7 7/16		.8698	10 7/16
1/8	.1250	1 1/2	3/8	.3750	4 1/2	5/8	.6250	7 1/2	7/8	.8750	10 1/2
	.1302	1 9/16		.3802	4 9/16		.6302	7 9/16		.8802	10 9/16
	.1354	1 5/8		.3854	4 5/8		.6354	7 5/8		.8854	10 5/8
9/64	.140625	1 11/16	25/64	.390625	4 11/16	41/64	.640625	7 11/16	57/64	.890625	10 11/16
	.1458	1 3/4		.3958	4 3/4		.6458	7 3/4		.8958	10 3/4
	.1510	1 13/16		.4010	4 13/16		.6510	7 13/16		.9010	10 13/16
5/32	.15625	1 7/8	13/32	.40625	4 7/8	21/32	.65625	7 7/8	29/32	.90625	10 7/8
	.1615	1 15/16		.4115	4 15/16		.6615	7 15/16		.9115	10 15/16
	.1667	2		.4167	5		.6667	8		.9167	11
11/64	.171875	2 1/16	27/64	.421875	5 1/16	43/64	.671875	8 1/16	59/64	.921875	11 1/16
	.1771	2 1/8		.4271	5 1/8		.6771	8 1/8		.9271	11 1/8
	.1823	2 3/16		.4323	5 3/16		.6823	8 3/16		.9323	11 3/16
3/16	.1875	2 1/4	7/16	.4375	5 1/4	11/16	.6875	8 1/4	15/16	.9375	11 1/4
	.1927	2 5/16		.4427	5 5/16		.6927	8 5/16		.9427	11 5/16
	.1979	2 3/8		.4479	5 3/8		.6979	8 3/8		.9479	11 3/8
13/64	.203125	2 7/16	29/64	.453125	5 7/16	45/64	.703125	8 7/16	61/64	.953125	11 7/16
	.2083	2 1/2		.4583	5 1/2		.7083	8 1/2		.9583	11 1/2
	.2135	2 9/16		.4635	5 9/16		.7135	8 9/16		.9635	11 9/16
7/32	.21875	2 5/8	15/32	.46875	5 5/8	23/32	.71875	8 5/8	31/32	.96875	11 5/8
	.2240	2 11/16		.4740	5 11/16		.7240	8 11/16		.9740	11 11/16
	.2292	2 3/4		.4792	5 3/4		.7292	8 3/4		.9792	11 3/4
15/64	.234375	2 13/16	31/64	.484375	5 13/16	47/64	.734375	8 13/16	63/64	.984375	11 13/16
	.2396	2 7/8		.4896	5 7/8		.7396	8 7/8		.9896	11 7/8
	.2448	2 15/16		.4948	5 15/16		.7448	8 15/16		.9948	11 15/16
1/4	.2500	3	1/2	.5000	6	3/4	.7500	9	1	1.0000	12

Table 4. Wire and Sheet Metal Gages

(Values in approximate decimals of an inch. As a number of gages are in use for various shapes and metals, it is advisable to state the thickness in thousandths when specifying gage number.)

Gage number	American (A.W.G.) or Brown & Sharpe (B. & S.) (for nonferrous wire and sheet)*	U. S. Steel Wire (Stl. W.G.) or Washburn & Moen or Roebling or Am. Steel & Wire Co. [A. (steel) W.G.] (for steel wire)	Birmingham (B.W.G.) (for steel wire) or Stubs Iron Wire (for iron or brass wire)†	Stubs (steel wire)	U. S. Standard (for sheet and plate metal, iron and steel)	Trenton Iron Co.	Standard Birmingham (B.G.) (for sheet and hoop metal)	British Imperial Standard Wire (S.W.G.) or New British Standard (N.B.S.) or English Legal Standard or Imperial Wire Gage	Gage number
0000000	—	0.4900	—	—	0.500	—	—	0.500	0000000
000000	—	.4615	—	—	.469	—	—	.464	000000
00000	—	.4305	—	—	.438	0.450	—	.432	00000
0000	0.460	.3938	0.454	—	.406	.400	—	.400	0000
000	.410	.3625	.425	—	.375	.360	0.5000	.372	000
00	.365	.3310	.380	—	.344	.330	.4452	.348	00
0	.325	.3065	.340	—	.312	.305	.3964	.324	0
1	.289	.2830	.300	0.227	.281	.285	.3532	.300	1
2	.258	.2625	.284	.219	.266	.265	.3147	.276	2
3	.229	.2437	.259	.212	.250	.245	.2804	.252	3
4	.204	.2253	.238	.207	.234	.225	.2500	.232	4
5	.182	.2070	.220	.204	.219	.205	.2225	.212	5
6	.162	.1920	.203	.201	.203	.190	.1981	.192	6
7	.144	.1770	.180	.199	.188	.175	.1764	.176	7
8	.128	.1620	.165	.197	.172	.160	.1570	.160	8
9	.114	.1483	.148	.194	.156	.145	.1398	.144	9
10	.102	.1350	.134	.191	.141	.130	.1250	.128	10
11	.091	.1205	.120	.188	.125	.1175	.1113	.116	11
12	.081	.1055	.109	.185	.109	.105	.0991	.104	12
13	.072	.0915	.095	.182	.094	.0925	.0882	.092	13
14	.064	.0800	.083	.180	.078	.080	.0785	.080	14
15	.057	.0720	.072	.178	.070	.070	.0699	.072	15
16	.051	.0625	.065	.175	.062	.061	.0625	.064	16
17	.045	.0540	.058	.172	.056	.0525	.0556	.056	17
18	.040	.0475	.049	.168	.050	.045	.0495	.048	18
19	.036	.0410	.042	.164	.0438	.040	.0440	.040	19
20	.032	.0348	.035	.161	.0375	.035	.0392	.036	20
21	.0285	.0317	.032	.157	.0344	.031	.0349	.032	21
22	.0253	.0286	.028	.155	.0312	.028	.0313	.028	22
23	.0226	.0258	.025	.153	.0281	.025	.0278	.024	23
24	.0201	.0230	.022	.151	.0250	.0225	.0248	.022	24
25	.0179	.0204	.020	.148	.0219	.020	.0220	.020	25
26	.0159	.0181	.018	.146	.0188	.018	.0196	.018	26
27	.0142	.0173	.016	.143	.0172	.017	.0175	.0164	27
28	.0126	.0162	.014	.139	.0156	.016	.0156	.0148	28
29	.0113	.0150	.013	.134	.0141	.015	.0139	.0136	29
30	.0100	.0140	.012	.127	.0125	.014	.0123	.0124	30
31	.0089	.0132	.010	.120	.0109	.013	.0110	.0116	31
32	.0080	.0128	.009	.115	.0102	.012	.0098	.0108	32
33	.0071	.0118	.008	.112	.0094	.011	.0087	.0100	33
34	.0063	.0104	.007	.110	.0086	.010	.0077	.0092	34
35	.0056	.0095	.005	.108	.0078	.0095	.0069	.0084	35
36	.0050	.0090	.004	.106	.0070	.009	.0061	.0076	36
37	.0045	.0085	—	.103	.0066	.0085	.0054	.0068	37
38	.0040	.0080	—	.101	.0062	.008	.0048	.0060	38
39	.0035	.0075	—	.099	—	.0075	—	.0052	39
40	.0031	.0070	—	.097	—	.007	—	.0048	40
41	—	.0066	—	.095	—	—	—	.0044	41
42	—	.0062	—	.092	—	—	—	.0040	42
43	—	.0060	—	.088	—	—	—	.0036	43
44	—	.0058	—	.085	—	—	—	.0032	44
45	—	.0055	—	.081	—	—	—	.0028	45
46	—	.0052	—	.079	—	—	—	.0024	46
47	—	.0050	—	.077	—	—	—	.0020	47
48	—	.0048	—	.075	—	—	—	.0016	48
49	—	.0046	—	.072	—	—	—	.0012	49
50	—	.0044	—	.069	—	—	—	.0010	50

Metric wire gage is ten times the diameter in millimeters.

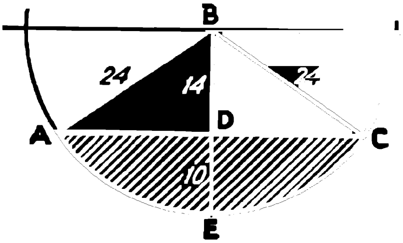
* Sometimes used for iron wire.

† Sometimes used for copper plate and for steel plate 12 gage and heavier and for steel tubes.

Table 5. Capacities of Horizontal Cylindrical Tanks

Method of Calculation

1. *Flat Ends.* Assume diameter=48 in. and length=120 in. Required the volume (in U. S. gallons) of shaded area of figure (*ACE*) of height 10 in.



Area ACE = area $ABCE$ − area ABC.

Area $ABCE = \left(\dfrac{2\angle ABD}{360}\right) \times$ area of circle, and $\angle ABD$ is found from its cosine which is 14/24.

$$\therefore\ \angle ABD = 54.25^\circ$$

$$\text{Area } ABC = 14 \times 24 \times \sin \angle ABD$$

or

$$14 \times 24 \times \sin 54.25 = 14 \times 24 \times .8116 = 272.7$$

$$\text{Area } ABCE = \frac{108.50}{360} \times \pi \times (24)^2 = 561.$$

$$\therefore\ \text{Area } ACE = 561 - 272.7 = 288.3$$

$$\therefore\ \text{Vol (U. S. gallons) per foot of length} = \frac{288.3 \times 12}{231} = 15$$

See Table 6.

To calculate the contents of a tank over one-half full, let *h* be the depth of the unfilled part of the tank. Find the volume corresponding to this depth from the table and subtract it from the volume of the full tank as given in the last column of each horizontal line of the table.

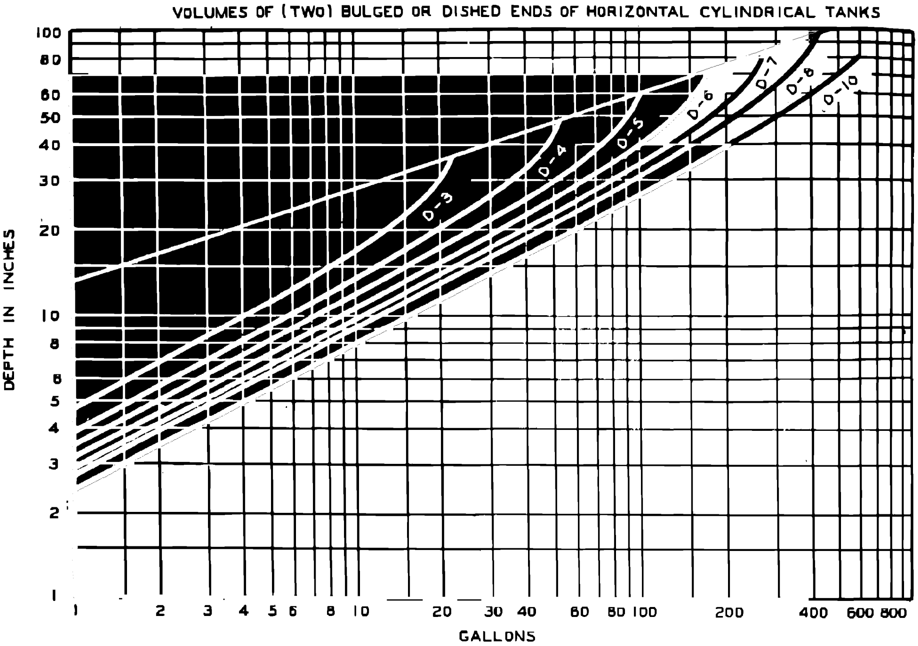
Table 6. Capacities of Horizontal Cylindrical Tanks in U. S. Gallons

Contents given for 1 ft of length

Flat ends

Diameter of tank, in.	Depth of liquid, in.																
	3	6	9	12	15	18	21	24	27	30	33	36	39	42	45	48	51
12	1.15	2.94	4.73	5.88													
18	1.45	3.86	6.61	9.36	11.77	13.22											
24	1.70	4.60	8.05	11.75	15.45	18.90	21.80	23.50									
30	1.91	5.23	9.27	13.72	18.36	23.00	27.45	31.49	34.81	36.72							
36	2.12	5.79	10.34	15.43	20.85	26.44	32.03	37.45	42.54	47.09	50.76	52.88					
42	2.28	6.31	11.31	16.97	23.07	29.46	35.99	42.51	48.90	55.00	60.65	65.66	69.69	71.97			
48	2.45	6.78	12.20	18.38	25.10	32.20	39.54	47.01	54.47	61.81	68.91	75.63	81.81	87.23	91.56	94.01	
54	2.60	7.22	13.04	19.68	26.97	34.72	42.80	51.08	59.49	67.90	76.18	84.26	92.01	99.30	105.94	111.76	116.38
60	2.75	7.64	13.82	20.91	28.72	37.06	45.82	54.88	64.11	73.45	82.78	92.01	101.07	109.83	118.17	125.98	133.07
66	2.89	8.04	14.56	22.07	30.37	39.28	48.65	58.39	68.42	78.59	88.87	99.14	109.31	119.34	129.08	138.45	147.36
72	3.02	8.42	15.26	23.17	31.92	41.36	51.32	61.71	72.45	83.41	94.54	105.76	116.98	128.11	139.07	149.81	160.20
78	3.15	8.78	15.94	24.21	33.41	43.34	53.85	64.87	76.27	87.97	99.90	111.97	124.12	136.27	148.34	160.27	171.97
84	3.26	9.12	16.57	25.24	34.85	45.24	56.29	67.87	79.91	92.30	104.98	117.85	130.87	143.95	157.03	170.05	182.92
90	3.43	9.46	17.20	26.20	36.21	47.05	58.61	70.75	83.39	96.43	109.81	123.46	137.28	151.22	165.25	179.27	193.21
96	3.50	9.79	17.80	27.13	37.52	48.81	60.84	73.52	86.73	100.39	114.44	128.79	143.40	158.17	173.07	188.01	202.95
102	3.61	10.10	18.37	28.01	39.00	50.49	62.99	76.18	89.94	104.20	118.89	133.92	149.25	164.81	180.53	196.36	212.25
108	3.71	10.39	18.94	28.90	40.03	52.14	65.09	78.74	93.04	107.87	123.17	138.87	154.89	171.19	187.71	204.37	221.15

Diameter of tank, in.	Depth of liquid, in.																
	54	57	60	63	66	69	72	75	78	81	84	87	90	93	96	102	108
12																	
18																	
24																	
30																	
36																	
42																	
48																	
54	118.98																
60	139.25	144.14	146.89														
66	155.66	163.17	169.69	174.84	177.73												
72	170.16	179.60	188.35	196.26	203.10	208.50	211.52										
78	183.37	194.38	204.90	214.83	224.03	232.30	239.46	245.09	248.24								
84	195.60	207.99	220.03	231.61	242.66	253.05	262.66	271.33	278.78	284.64	287.90						
90	207.03	220.68	234.06	247.10	259.74	271.88	283.44	294.28	304.29	313.29	321.03	327.06	330.49				
96	217.85	232.62	247.23	261.58	275.63	289.29	302.50	315.18	327.21	338.50	348.89	358.22	366.23	372.52	376.02		
102	228.12	243.95	259.67	275.23	290.56	305.59	320.28	334.54	348.30	361.49	373.99	385.48	396.47	406.11	414.38	424.48	
108	238.05	254.75	271.53	288.19	304.71	321.01	337.03	352.73	368.03	382.86	397.16	410.81	423.76	435.87	447.00	465.51	475.90


VOLUMES OF (TWO) BULGED OR DISHED ENDS OF HORIZONTAL CYLINDRICAL TANKS

Constructed from Doolittle Formula. Ind. Eng. Chem., March, 1928.
V (both ends) = $0.0000328h^2(3r - h)$ gallons.
h = depth in inches.
r = radius of tank in inches.
In curves D = diameter of tank in feet.

Table 7. Greek Alphabet

Greek letter	Greek name	English equivalent	Greek letter	Greek name	English equivalent
Α α	Alpha	a	Ν ν	Nu	n
Β β	Beta	b	Ξ ξ	Xi	x
Γ γ	Gamma	g	Ο ο	Omicron	ŏ
Δ δ	Delta	d	Π π	Pi	p
Ε ε	Epsilon	ĕ	Ρ ρ	Rho	r
Ζ ζ	Zeta	z	Σ σ	Sigma	s
Η η	Eta	ē	Τ τ	Tau	t
Θ θ	Theta	th	Υ υ	Upsilon	u
Ι ι	Iota	i	Φ φ	Phi	ph
Κ κ	Kappa	k	Χ χ	Chi	ch
Λ λ	Lambda	l	Ψ ψ	Psi	ps
Μ μ	Mu	m	Ω ω	Omega	ō

14. CONVERSION TABLES

Equivalents of Measure

Use of all tables. To convert a known value to a desired measure, multiply by the known units equivalent.

Example: The wall of a room is given as 20.25 m; its length therefore must be 3.280333 × 20.25 = 66.42674 ft.

Notations: The following expressions will be used: $\overset{2}{0}\ \overset{3}{0}\ \overset{4}{0}\ \overset{5}{0}$, etc., indicating the number of ciphers to be used after the decimal point.

Example: 1 sq rod $= 0.\overset{5}{0}9766 = 0.000009766$ sq miles, or 1 grain $= 0.\overset{2}{0}2083 = 0.002083$ oz troy. 1 grain $= 0.\overset{4}{0}6480 = 0.00006480$ kg, or 1 kg/m^3 $= 0.\overset{4}{0}3613 = 0.00003613$ lb/in^3.

Lengths

1 meter (m) = 10 decimeters (dm) = 100 centimeters (cm) = 1000 millimeters (mm) = 0.001 kilometer (km)
To convert metric units merely adjust the decimal point.
1 meter (m) = 39.37 in. (U. S. Standard) = 39.370113 in. (British Standard)
1 millimeter (mm) = 1000 microns (μ) = 0.03937 in. = 39.37 mils.
1 yard (U. S. Standard) = 1.0000029 yards (British standard)

Meters m	Inches in.	Feet ft	Yard yd	Rods 4	Miles Statute	 U. S. Nautical	Kilometers km
1	39.37	3.280833	1.093611	0.198838	$0.\overset{3}{0}6214$	$0.\overset{3}{0}5396$	0.001
0.02540	1	0.08333	0.02778	$0.\overset{2}{0}5051$	$0.\overset{4}{0}1578$	$0.\overset{4}{0}1371$	$0.\overset{4}{0}2540$
0.30480	12	1	0.33333	0.06061	$0.\overset{3}{0}1894$	$0.\overset{3}{0}1645$	$0.\overset{3}{0}3048$
0.91440	36	3	1	0.18182	$0.\overset{3}{0}5682$	$0.\overset{3}{0}4934$	$0.\overset{3}{0}9144$
5.02921	198	16.5	5.5	1	$0.\overset{2}{0}3125$	$0.\overset{2}{0}2714$	$0.\overset{2}{0}5029$
1,609.35	63,360	5,280	1,760	320	1	0.86839	1.60935
1,853.25	72,962.5	6,080.20	2,026.73	368.497	1.15155	1	1.85325
1,000	39,370	3,280.83	1,093.61	198.838	0.62137	0.53959	1

Surfaces and Areas

1 sq meter (m^2) = 100 sq decimeters (dm^2) = 10,000 sq centimeters (cm^2) = 0.0001 hectare (ha)
1 sq millimeter (mm^2) = 0.01 cm^2 = 0.00155 sq in. = 1,973.5 circular mils

Sq Meters m^2	Sq Inches sq in.	Sq Feet sq ft	Sq Yards sq yd	Acres A	Hectares ha	Sq Miles statute	Sq Kilometers km^2
				3		6	5
1	1549.99	10.7639	1.19599	0.02471	0.0001	0.03861	0.01
3		2	3	6	7	9	9
0.06452	1	0.06944	0.07716	0.01594	0.06452	0.02491	0.06452
				4	5	7	7
0.09290	144	1	0.11111	0.02296	0.09290	0.03587	0.09290
				3	4	6	6
0.83613	1,296	9	1	0.02066	0.08361	0.03228	0.08361
					2	5	4
						2	2
4,046.87	6,272,640	43,560	4,840	1	0.40469	0.01563	0.04047
						2	
10,000	15,499,969	107,639	11,959.9	2.47104	1	0.03861	0.01
2,589,999		27,878,400	3,097,600	640	259.000	1	2.59000
1,000,000		10,763,867	1,195,985	247.104	100	0.38610	1

1 sq rod, sq pole, or sq perch = 625 sq links = $\frac{1}{160}$ acre.

Volume and Capacity

1 cu meter (m^3) = 1,000 cu decimeters (dm^3) = 1,000,000 cu centimeters (cm^3)
1 liter (l) = 10 deciliters (dl) = 100 centiliters (cl) = 1,000 milliliters (ml) = 1,000 cu centimeters (cm^3 or cc)
1 liter (l) = 0.1 decaliter (dkl) = 0.01 hectoliter (hl) = 1 cu decimeter (dm^3)

Cubic Decimeters dm^3 l	Cubic Inches cu in.	Cubic Feet cu ft	Cubic Yard cu yd	U. S. Quarts		U. S. Gallons		U. S. Bushels bu
				liquid l qt	dry d qt	liquid l gal	dry d gal	
			2					
1	61.0234	0.035314	0.01308	1.05668	0.90808	0.264178	0.22702	0.02838
		3	4			2	2	3
0.01639	1	0.05787	0.02143	0.01732	0.01488	0.04329	0.03720	0.04650
28.3170	1,728	1	0.03704	29.9221	25.7140	7.48055	6.42851	0.80356
764.559	46,656	27	1	807.896	694.279	201.974	173.570	21.6962
			2					
0.94636	57.75	0.03342	0.01238	1	0.85937	0.25	0.21484	0.02686
			2					
1.10123	67.2006	0.03889	0.01440	1.16365	1	0.29091	0.25	0.03125
			2					
3.78543	231	0.13368	0.04951	4	3.43747	1	0.85937	0.10742
			2					
4.40492	268.803	0.15556	0.05761	4.65460	4	1.16365	1	0.125
35.2393	2,150.42	1.24446	0.04609	37.2368	32	9.30920	8	1
4.54374	277.274	0.160459	0.0059429	4.30128	4.1267	1.20032	1.0317	0.12896

Bottom row lists the equivalents for one imperial gallon.

U. S. dry measure: 1 bushel = 4 pecks = 8 gallons = 32 quarts = 64 pints.
U. S. liquid measure: 1 gallon = 4 quarts = 8 pints = 32 gills = 128 fluid ounces.
U. S. Apoth. measure: 1 fl. ounce (f℥) = 8 fl. drams (fʒ) = 480 minims (♏) = 29.574 cm^3.

Example: For South America a pump has to be furnished of 1,000 liters per minute capacity. According to American practice this would be 1,000 × 0.264178 = 264.178 gpm.

Masses and Weights

1 gram (g) = 10 decigrams (dg) = 100 centigrams (cg) = 1,000 milligrams (mg) = 0.001 kilograms (kg)
1 kilogram (kg) = 1 liter or cu decimeter of water at 4 degrees C, 45° Lat. and sea level = 15,432.35639 grains.
1 U. S. liquid gal = 8.34545 lb, 1 Imperial gal = 10.0221 lb, 1 cu ft = 62.4283 lb water as above.

Kilograms kg	Grains gr	Ounces		Pounds		Tons		
		Troy oz t	Avoir oz av	Troy lb t	Avoir lb av	Net-short 2,000 lb	Gross-long 2,240 lb	Metric 1,000 kg
						2	3	
1	1,5432.4	32.1507	35.2740	2.67923	2.20462	0.01102	0.09842	0.001
4		2	2	3	3	7	7	7
0.06480	1	0.02083	0.02286	0.01736	0.01429	0.07143	0.06378	0.06480
						4	4	4
0.03110	480	1	1.09714	0.08333	0.06857	0.03429	0.03061	0.03110
						4	4	4
0.02835	473.5	0.91146	1	0.07595	0.06250	0.03125	0.02790	0.02835
						3	3	3
0.37324	5,760	12	13.1657	1	0.82286	0.04114	0.03674	0.03732
							3	3
0.45359	7,000	14.5833	16	1.21528	1	0.00050	0.04464	0.04536
907.185	14,000,000	29,166.7	32,000	2,430.56	2,000	1	0.89286	0.90719
1,016.05	15,680,000	32,666.7	35,840	2,722.22	2,240	1.12	1	1.01605
1,000	15,432,356	32,150.7	35,274.0	2,679.23	2,204.62	1.10231	0.98421	1

The pound, ounce and grain Apothecary have the same weight as those of Troy weight.
1 pound (Apoth.) = 12 ounces (℥) = 96 drams (ʒ) = 288 scruples (℈) = 5760 grains (gr)
1 pound (Troy) = 12 ounces (℥) = 240 pennyweight (dwt) = 5760 grains (gr)
1 hundredweight = 1/20 long ton = 4 quarters = 8 stone = 112 lbs = 50.8024 kg
Example: To make 10 metric tons of ice per day would require a plant with a capacity of 10 × 1.10231 = 11.023 ton American practice. When capacities in long tons are specified, then a 10-gross-ton plant requires 10 × 1.12 = 11.2 U. S. tons ice-making capacity.

Pressure

1 dyne per sq centimeter = 0.00101979 g/cm^2 = 0.000466646 poundals/in^2.
1 gram per sq centimeter = 980.5966 dynes/cm^2 = 0.457592 poundals/in^2.
1 poundal per sq in. = 2,142.95 dynes/cm^2 = 2.18536 g/cm^2 = 0.0310832 pound/in^2.

Kilograms per sq centimeter kg/cm^2	Pounds per sq inch psi	Pounds per sq foot psf	Atmospheres, Standard 760 mm	Columns of Mercury Hg. 13.59593 sp g		Columns of Water, max density at 4C, 39F	
				Millimeters	Inches	Meters	Feet
1	14.2234	2,048.17	0.96778	735.514	28.9572	10	32.8083
0.07031	1	144	0.06804	51.7116	2.03588	0.70307	2.30665
3	2		3			2	
0.04882	0.06944	1	0.04725	0.35911	0.01414	0.04882	0.01602
1.03329	14.6969	2,116.35	1	760	29.9212	10.3329	33.9006
2			2				
0.01360	0.01934	2.78468	0.01316	1	0.03937	0.01360	0.04461
0.03453	0.49119	70.7310	0.03342	25.4001	1	0.34534	1.13299
0.10	1.42234	204.817	0.09678	73.5514	2.89572	1	3.28083
0.03048	0.43353	62.4283	0.02950	22.4185	0.88262	0.30480	1

Density—Specific Weights

1 dyne per cu centimeter = 0.00101979 gram/cm³ = 0.00118528 poundals/in³.
1 gram per cu centimeter = 980.5966 dynes/cm³ = 1.162283 poundals/in³.
1 poundal per cu in. = 843.683 dynes/cm³ = 0.860378 g/cm³ = 0.0310832 pound/in³.

Grams per cu centimeter g/cm³	Pounds per cu inch lb/in³	Pounds per cu foot lb/ft³	Pounds per cu yard lb/yd³	Kilograms per cu meter kg/m³	Pounds per bushel U. S.	Pounds per gallon, dry, U. S.	Pounds per gallon, liquid, U. S.
1	0.03613	62.4283	1,685.56	1,000	77.6893	9.71116	8.34545
27.6797	1	1,728	46,656	2,7679.7	2,150.42	268.803	231
0.01602	$0.0_{3}5787$	1	27	16.0184	1.24446	0.15556	0.13368
$0.0_{3}5933$	$0.0_{4}2143$	0.03704	1	0.59327	0.04609	$0.0_{2}5762$	$0.0_{2}4951$
0.001	$0.0_{4}3613$	0.06243	1.68556	1	0.07769	$0.0_{2}9711$	$0.0_{2}8345$
0.01287	$0.0_{3}4650$	0.80356	21.6962	12.8718	1	0.125	0.10742
0.010297	$0.0_{2}3720$	6.42851	173.570	102.974	8	1	0.85937
0.11983	$0.0_{2}4329$	7.48052	201.974	119.826	9.30920	1.16365	1
	3						

The specific gravity of a substance is the ratio of its density to that of water at 4C, both densities being expressed in the same units.

The density of a substance is its mass per unit volume.

The specific weight of a substance is its weight per unit volume.

Heat, Energy, Work

1 dyne-centimeter = 1 erg = 0.00101979 gram-centimeter = $0.0_{7}737612$ foot-pound.
1 gram-centimeter = 980.5966 ergs = $0.0_{4}7233$ foot-pounds.
1 foot-pound, ft-lb = 13,557,300 ergs = 13,825.5 gram-centimeter.

Kilogram meters kg-m	Foot pounds ft-lb	Horsepower—Hours U. S. hp-hr	Horsepower—Hours Metric 75 kg-m-hr	Kilowatt hours kw-hr	Joules 10⁷ ergs j	Thermal Units Btu	Thermal Units kg-cal
1	7.23300	$0.0_{5}3653$	$0.0_{5}3704$	$0.0_{5}2724$	9.80597	$0.0_{2}9296$	$0.0_{2}2342$
0.13826	1	$0.0_{6}5051$	$0.0_{6}5121$	$0.0_{6}3766$	1.35573	$0.0_{2}1285$	$0.0_{3}3239$
273,745	1,980,000	1	1.01387	0.74565	2,684,340	2,544.65	641.240
270,000	1,952,910	0.98632	1	0.73545	2,647,610	2,509.83	632.467
367,123	2,655,403	1.34111	1.35972	1	3,600,000	3,412.66	859.975
0.10198	0.73761	$0.0_{6}3725$	$0.0_{6}3777$	$0.0_{6}2778$	1	$0.0_{3}9480$	$0.0_{3}2389$
107.577	778.104	$0.0_{3}3930$	$0.0_{3}3984$	$0.0_{3}2930$	1,054.90	1	0.25200
426.900	3,087.77	$0.0_{2}1559$	$0.0_{2}1581$	$0.0_{2}1163$	4,186.17	3.96832	1

Power, Rate of Energy and Heat

1 erg per sec = 1 dyne-cm/sec = $0.\overset{7}{0}737612$ foot-pound/sec.
1 gram-centimeter per second = 980.5966 ergs/sec = $0.\overset{4}{0}7238$ foot-pound/sec.
1 foot-pound per second = 13,557,300 ergs/sec = 13,825.5 gram-cm/sec.

Kilogram Meters per second kg-m/sec	Foot-pounds per second ft-lb/sec	Horsepower		Kilowatt kw	Watts 10^7 ergs/sec	Thermal Units per Sec	
		U. S. 550 ft-lb/sec	Metric 75 kg-m/sec			Btu/sec	kg-cal/sec
1	7.23300	0.01315	0.01333	$0.\overset{2}{0}9806$	9.80597	$0.\overset{2}{0}9296$	$0.\overset{2}{0}2342$
0.13826	1	$0.\overset{2}{0}1818$	$0.\overset{2}{0}1843$	$0.\overset{2}{0}1356$	1.35573	$0.\overset{2}{0}1285$	$0.\overset{3}{0}3237$
76.0404	550	1	1.01387	0.74565	745.650	0.70685	0.17812
75	542.475	0.98632	1	0.73545	735.448	0.69718	0.17569
100	723.300	1.31509	1.33333	0.98060	980.597	0.92957	0.23425
101.979	737.612	1.34111	1.35972	1	1,000	0.94796	0.23888
0.10198	0.73761	$0.\overset{2}{0}1341$	$0.\overset{2}{0}1360$	0.001	1	$0.\overset{3}{0}9480$	$0.\overset{3}{0}2389$
107.577	778.104	1.41474	1.43436	1.05490	1,054.90	1	0.25200
426.900	3,087.77	5.61412	5.69200	4.18617	4,186.17	3.96832	1

The ton refrigeration is 288,000 Btu/24 hr = 12,000 Btu/hr = 200 Btu/min = 3.3333 Btu/sec.

Example: To absorb the heat developed in expending the energy of one hp/sec, $\frac{0.70685}{3.3333} = 0.21206$ ton refrigeration capacity is required.

Velocities and Accelerations

1 centimeter per sec = 0.0328083 ft per sec.
1 radian per second = 57.2958 degrees per sec = 0.159155 revolution per sec.
1 gravity = 980.5966 centimeters per sec per sec = 32.1717 ft per sec per sec.

Meters per second mps	Feet per second fps	Miles per hour mph	Knots U. S.	Kilometers per hour	Meters per sec/sec mps^2	Feet per sec/sec fps^2	Miles per hour/sec mph-s
1	3.28083	2.23693	1.94254	3.6	1	0.28083	2.23693
0.30480	1	0.68182	0.59209	1.09728	0.30480	1	0.68182
0.44704	1.46667	1	0.86839	1.60935	0.44704	1.46667	1
0.51479	1.68894	1.15155	1	1.85325	0.27778*	0.91134*	0.62137*
0.27778	0.91134	0.62137	0.53959	1			

* Equivalents for 1 Kilometer per hour per sec.

Temperature Conversion Table

Deg F	Deg C	Deg F	Deg C	Deg F	Deg C	Deg F	Deg C
-120.	-84.45	-77.80	-61.	-35.	-37.22	8.	-13.33
-119.20	-84.	-77.	-60.56	-34.60	-37.	8.60	-13.
-119.	-83.89	-76.	-60.	-34.	-36.67	9.	-12.78
-118.	-83.33	-75.	-59.44	-33.	-36.11	10.	-12.22
-117.40	-83.	-74.20	-59.	-32.80	-36.	10.40	-12.
-117.	-82.78	-74.	-58.89	-32.	-35.56	11.	-11.67
-116.	-82.22	-73.	-58.33	-31.	-35.	12.	-11.11
-115.60	-82.	-72.40	-58.	-30.	-34.44	12.20	-11.
-115.	-81.67	-72.	-57.78	-29.20	-34.	13.	-10.56
-114.	-81.11	-71.	-57.22	-29.	-33.89	14.	-10.
-113.80	-81.	-70.60	-57.	-28.	-33.33	15.	- 9.44
-113.	-80.56	-70.	-56.67	-27.40	-33.	15.80	- 9.
-112.	-80.			-27.	-32.78	16.	- 8.89
-111.	-79.45	-69.	-56.11	-26.	-32.22	17.	- 8.33
-110.20	-79.	-68.80	-56.	-25.60	-32.	17.60	- 8.
-110.	-78.89	-68.	-55.56	-25.	-31.67	18.	- 7.78
-109.	-78.33	-67.	-55.	-24.	-31.11	19.	- 7.22
-108.40	-78.	-66.	-54.44	-23.80	-31.	19.40	- 7.
-108.	-77.78	-65.20	-54.	-23.	-30.56	20.	- 6.67
-107.	-77.11	-65.	-53.89	-22.	-30.	21.	- 6.11
-106.60	-77.	-64.	-53.33	-21.	-29.44	21.20	- 6.
-106.	-76.67	-63.40	-53.	-20.20	-29.	22.	- 5.56
-105.	-76.11	-63.	-52.78	-20.	-28.89	23.	- 5.
-104.80	-76.	-62.	-52.22	-19.	-28.33	24.	- 4.44
-104.	-75.56	-61.60	-52.	-18.40	-28.	24.80	- 4.
-103.	-75.	-61.	-51.67	-18.	-27.78	25.	- 3.89
-102.	-74.45	-60.	-51.11	-17.	-27.22	26.	- 3.33
-101.20	-74.	-59.80	-51.	-16.60	-27.	26.60	- 3.
-101.	-73.89	-59.	-50.56	-16.	-26.67	27.	- 2.78
-100.	-77.33	-58.	-50.	-15.	-26.11	28.	- 2.22
- 99.40	-73.	-57.	-49.44	-14.80	-26.	28.40	- 2.
- 99.	-72.78	-56.20	-49.	-14.	-25.56	29.	- 1.67
- 98.	-72.22	-56.	-48.89	-13.	-25.	30.	- 1.11
- 97.60	-72.	-55.	-48.33	-12.	-24.44	30.20	- 1.
- 97.	-71.67	-54.40	-48.	-11.20	-24.	31.	- 0.56
- 96.	-71.11	-54.	-47.78	-11.	-23.89	32.	0.
- 95.80	-71.	-53.	-47.22	-10.	-23.33	33.	0.56
- 95.	-70.56	-52.60	-47.	- 9.40	-23.	33.80	1.
- 94.	-70.	-52.	-46.67	- 9.	-22.78	34.	1.11
- 93.	-69.45	-51.	-46.11	- 8.	-22.22	35.	1.67
- 92.20	-69.	-50.80	-46.	- 7.60	-22.	35.60	2
- 92.	-68.89	-50.	-45.56	- 7.	-21.67	36.	2.22
- 91.	-68.33	-49.	-45.	- 6.	-21.11	37.	2.78
- 90.40	-68.	-48.	-44.44	- 5.80	-21.	37.40	3.
-90.	-67.78	-47.20	-44.	- 5.	-20.56	38.	3.33
-89.	-67.22	-47.	-43.89	- 4.	-20.	39.	3.89
-88.60	-67.	-46.	-43.33	- 3.	-19.44	39.20	4.
-88.	-66.67	-45.30	-43.	- 2.20	-19.	40.	4.44
-87.	-66.11	-45.	-42.78	- 2.	-18.89	41.	5.
-86.80	-66.	-44.	-42.22	- 1.	-18.33	42.	5.56
-86.	-65.56	-43.60	-42.	- 0.40	-18.	42.80	6.
-85.	-65.	-43.	-41.67	0	-17.78	43.	6.11
-84.	-64.44	-42.	-41.11	.1	-17.22	44.	6.67
-83.20	-64.	-41.80	-41.	1.40	-17.	44.60	7.
-83.	-63.89	-41.	-40.56	2.	-16.67	45.	7.22
-82.	-63.33	-40.	-40.	3.	-16.11	46.	7.78
-81.40	-63.	-39.	-39.44	3.20	-16.	46.40	8.
-81.	-62.78	-38.20	-39.	4.	-15.56	47.	8.33
-80.	-62.22	-38.	-38.89	5.	-15.	48.	8.89
-79.60	-62.	-37.	-38.33	6.	-14.44	48.20	9.
-79.	-61.67	-36.40	-38.	6.80	-14.	49.	9.44
-78.	-61.11	-36.	-37.78	7.	-13.89	50.	10.

Temperature Conversion Table (Concluded)

Deg F	Deg C	Deg F	Deg C	Deg F	Deg C	Deg F	Deg C
51.	10.56	91.40	33.	132.80	56.	173.	78.33
51.80	11.	92.	33.33	133.	56.11	174.	78.89
52.	11.11	93.	33.89	134.	56.67	174.20	79.
53.	11.67	93.20	34.	134.60	57.	175.	79.44
53.60	12.	94.	34.44	135.	57.22	176.	80.
54.	12.22	95.	35.	136.	57.78	177.	80.56
55.	12.78	96.	35.56	136.40	58.	177.80	81.
55.40	13.	96.80	36.	137.	58.33	178.	81.11
56.	13.33	97.	36.11	138.	58.89	179.	81.67
57.	13.89	98.	36.67	138.20	59.	179.60	82.
57.20	14.	98.60	37.	139.	59.44	180.	82.22
58.	14.44	99.	37.22	140.	60.	181.	82.78
59.	15.	100.	37.78	141.	60.56	181.40	83.
60.	15.56	100.40	38.	141.80	61.	182.	83.33
60.80	16.	101.	38.33	142.	61.11	183.	83.89
61.	16.11	102.	38.89	143.	61.67	183.20	84.
62.	16.67	102.80	39.	143.60	62.	184.	84.44
62.60	17.	103.	39.44	144.	62.22	185.	85.
63.	17.22	104.	40.	145.	62.78	186.	85.56
64.	17.78	105.	40.56	145.40	63.	186.80	86.
64.40	18.	105.80	41.	146.	63.33	187.	86.11
65.	18.33	106.	41.11	147.	63.89	188.	86.67
66.	18.89	107.	41.67	147.20	64.	188.60	87.
66.20	19.	107.60	42.	148.	64.44	189.	87.22
67.	19.44	108.	42.22	149.	65.	190.	87.78
68.	20.	109.	42.78	150.	65.56	190.40	88.
69.	20.56	109.40	43.	150.80	66.	191.	88.33
69.80	21.	110.	43.33	151.	66.11	192.	88.89
70.	21.11	111.	43.89	152.	66.67	192.20	89.
71.	21.67	111.20	44.	152.60	67.	193.	89.44
71.60	22.	112.	44.44	153.	67.22	194.	90.
72.	22.22	113.	45.	154.	67.78	195.	90.56
73.	22.78	114.	45.56	154.40	68.	195.80	91.
73.40	23.	114.80	46.	155.	68.33	196.	91.11
74.	23.33	115.	46.11	156.	68.89	197.	91.67
75.	23.89	116.	46.67	156.20	69.	197.60	92.
75.20	24.	116.60	47.	157.	69.44	198.	92.22
76.	24.44	117.	47.22	158.	70.	199.	92.78
77.	25.	118.	47.78	159.	70.56	199.40	93.
78.	25.56	118.40	48.	159.80	71.	200.	93.33
78.80	26.	119.	48.33	160.	71.11	201.	93.89
79.	26.11	120.	48.89	161.	71.67	201.20	94.
80.	26.67	120.20	49.	161.60	72.	202.	94.44
80.60	27.	121.	49.44	162.	72.22	203.	95.
81.	27.22	122.	50.	163.	72.78	204.	95.56
82.	27.78	123.	50.56	163.40	73.	204.80	96.
82.40	28.	123.80	51.	164.	73.33	205.	96.11
83.	28.33	124.	51.11	165.	73.89	206.	96.67
84.	28.89	125.	51.67	165.20	74.	206.60	97.
84.20	29.	125.60	52.	166.	74.44	207.	97.22
85.	29.44	126.	52.22	167.	75.	208.	97.78
86.	30.	127.	52.78	168.	75.56	208.40	98.
87.	30.56	127.40	53.	168.80	76.	209.	98.33
87.80	31.	128.	53.33	169.	76.11	210.	98.89
88.	31.11	129.	53.89	170.	76.67	210.20	99.
89.	31.67	129.20	54.	170.60	77.	211.	99.44
89.60	32.	130.	54.44	171.	77.22	212.	100.
90.	32.22	131.	55.	172.	77.78		
91.	32.78	132.	55.56	172.40	78.		

Deg C $=\frac{5}{9}$ (F $-$ 32);

Deg F $= 32 + \frac{9}{5}$ C

Viscosity*

Dynamic or Absolute Viscosity, μ (mu)

gm/cm sec or dyne-sec/cm^2 Poise	gm/cm sec or dyne-sec/cm^2 Centipoises	lb/ft sec or poundal sec/ft^2	Slug/ft sec lb$_f$ sec/ft^2	kg/m hr
1	100	0.0672	$0.0^{2}209$	360
14.88	1488	1	0.03105	5356
478.5	47,850	32.174	1	172,300
$0.0^{2}2778$	0.2778	$0.0^{3}1867$	$0.0^{5}5804$	1

Kinematic viscosity $\nu=\mu/\rho$ where μ is the dynamic or absolute viscosity and ρ is the density of the fluid. Kinematic viscosity is termed a centistoke in the metric system.

Relative viscosity is the ratio of dynamic viscosity of the fluid to that of water at a definite reference temperature, usually 20 C or 68 F, where the viscosity of water is practically 1/100 poise or 1 centipoise.

Specific viscosity: Saybolt is a very common method of measuring and stating viscosity and the results stated in Saybolt seconds. Since the rate of flow through the nozzle of the Saybolt viscometer depends on the dynamic viscosity and the head pressure, and since the pressure head is a function of the density of the fluid, Saybolt viscosity is a function of the kinematic viscosity.

$$\nu=0.0022\times t-\frac{1.8}{t}\text{ poise/gram,}$$

$$\nu=\left(0.0022\times t\times \rho-\frac{1.8}{t}\rho\right)\text{ poise,}$$

Also

$$\nu=0.00000237t-\frac{0.00194}{t}\text{ sq ft per sec}$$

Some commercial measurements of viscosity are made with the Engler viscometer and the results are stated in Engler seconds.

$$\nu=0.00147t-\frac{3.74}{t}\text{ poise/gram}$$

$$\nu=0.00147t\rho-3.74\frac{\rho}{t}\text{ poise}$$

where μ, ν and ρ are as above, and t is the time in Saybolt seconds or Engler seconds for the respective methods.

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* Prepared by R. J. Panlener.

Enthalpy

1 kg-cal/kg = 1.8 Btu/lb
1 Btu/lb = 0.555 kg-cal/kg

Entropy

1 kg-cal/kg deg C = 1 Btu/lb deg F

Thermal Conductivities

$\frac{\text{Btu ft}^*}{\text{hr ft}^2\text{ F}}$	$\frac{\text{Btu in.}}{\text{hr ft}^2\text{ F}}$	$\frac{\text{Chu in.}^\dagger}{\text{hr ft}^2\text{ C}}$	$\frac{\text{g cal cm}}{\text{sec cm}^2\text{ C}}$	$\frac{\text{kilo ergs}}{\text{sec cm}^2\text{ C}}$	$\frac{\text{w cm}}{\text{cm}^2\text{ C}}$	$\frac{\text{kg cal m}}{\text{hr m}^2\text{ C}}$
1.000	12.000	12.000	$0.0^{2}413$	173.	0.0173	1.49
0.0833	1.000	1.000	$0.0^{3}344$	14.4	$0.0^{2}144$	0.124
0.0833	1.000	1.000	$0.0^{3}344$	14.4	$0.0^{2}144$	0.124
242.	2,905.	2,905.	1.000	41,900.	4.19	361.
$0.0^{2}578$	0.0694	0.0694	$0.0^{4}239$	1.000	$0.0^{3}1$	$0.0^{2}861$
57.8	694.	694.	0.239	10,000.	1.000	86.1
0.671	8.05	8.05	$0.0^{2}277$	116.	0.0116	1.000

* Read: Btu foot per hour square foot degree Fahrenheit.

† The pound-centigrade unit, PCu, or centigrade heat unit Chu, is the heat required to raise one pound of water one degree centigrade and equals 1.8 Btu.

Coefficients of Heat Transfer

$\frac{\text{Btu}}{\text{hr ft}^2\text{ F}}$	$\frac{\text{kg-cal}}{\text{hr m}^2\text{ C}}$	$\frac{\text{Chu}}{\text{hr ft}^2\text{ C}}$	$\frac{\text{g-cal}}{\text{sec cm}^2\text{ C}}$	$\frac{\text{Watts}}{\text{cm}^2\text{ C}}$	$\frac{\text{Watts}}{\text{m}^2\text{ F}}$	$\frac{\text{hp}}{\text{ft}^2\text{ F}}$
1.000	4.88	1.000	$0.0^{3}1355$	$0.0^{3}568$	$0.0^{2}204$	$0.0^{3}394$
.205	1.000	.205	$0.0^{4}278$	$0.0^{3}116$	$0.0^{3}418$	$0.0^{4}807$
1.000	4.88	1.000	$0.0^{3}1355$	$0.0^{3}568$	$0.0^{2}204$	$0.0^{3}394$
7,380.	36,000.	7,380.	1.000	4.19	15.1	2.91
1,761.	8,592.	1,761.	0.239	1.000	3.697	0.694
490.	2,392.	490.	0.0664	0.278	1.000	0.193
2,538.	12,390.	2,538.	0.344	1.44	5.18	1.000

Absolute Pressure and Equivalent Vacuum Values

Inches of mercury at the density of 32 F

psia	in. Hg vac	psia	in. Hg vac	psia	in. Hg vac	psia	in. Hg vac
0	29.921	3.6	22.590	7.2	15.261	10.767	8.0
0.1	29.716	3.645	22.5	7.3	15.057	10.8	7.932
0.2	29.512	3.7	22.387	7.329	15.0	10.9	7.728
0.206	29.5	3.8	22.183	7.4	14.854	11.0	7.525
0.3	29.309	3.890	22.0	7.5	14.650	11.013	7.5
0.4	29.105	3.9	21.979	7.574	14.5	11.1	7.321
0.452	29.0	4.0	21.776	7.6	14.447	11.2	7.117
0.5	28.901	4.1	21.572	7.7	14.243	11.258	7.0
0.6	28.698	4.136	21.5	7.8	14.039	11.3	6.914
0.698	28.5	4.2	21.369	7.820	14.0	11.4	6.710
0.7	28.494	4.3	21.165	7.9	13.836	11.5	6.507
0.8	28.291	4.381	21.0	8.0	13.632	11.504	6.5
0.9	28.087	4.4	20.961	8.065	13.5	11.6	6.303
0.943	28.0	4.5	20.758	8.1	13.429	11.7	6.100
1.0	27.883	4.6	20.554	8.2	13.225	11.749	6.0
1.1	27.680	4.627	20.5	8.3	13.022	11.8	5.896
1.189	27.5	4.7	20.351	8.311	13.0	11.9	5.692
1.2	27.476	4.8	20.147	8.4	12.818	11.995	5.5
1.3	27.273	4.873	20.0	8.5	12.614	12.0	5.489
1.4	27.069	4.9	19.944	8.557	12.5	12.1	5.285
1.434	27.0	5.0	19.740	8.6	12.411	12.2	5.082
1.5	26.866	5.1	19.536	8.7	12.207	12.241	5.0
1.6	26.662	5.118	19.5	8.8	12.044	12.3	4.878
1.680	26.5	5.2	19.333	8.802	12.0	12.4	4.674
1.7	26.458	5.3	19.129	8.9	11.780	12.486	4.5
1.8	26.255	5.364	19.0	9.0	11.596	12.5	4.471
1.9	26.051	5.4	18.926	9.048	11.5	12.6	4.267
1.926	26.0	5.5	18.722	9.1	11.393	12.7	4.064
2.0	25.848	5.6	18.518	9.2	11.189	12.732	4.0
2.1	25.644	5.609	18.5	9.293	11.0	12.8	3.860
2.171	25.5	5.7	18.315	9.3	10.986	12.9	3.656
2.2	25.440	5.8	18.111	9.4	10.782	12.977	3.5
2.3	25.237	5.855	18.0	9.5	10.578	13.0	3.453
2.4	25.033	5.9	17.908	9.539	10.5	13.1	3.249
2.417	25.0	6.0	17.704	9.6	10.375	13.2	3.046
2.5	24.830	6.1	17.500	9.7	10.171	13.223	3.0
2.6	24.626	6.101	17.5	9.785	10.0	13.3	2.842
2.662	24.5	6.2	17.297	9.8	9.968	13.4	2.639
2.7	24.422	6.3	17.093	9.9	9.764	13.468	2.5
2.8	24.219	6.346	17.0	10.0	9.561	13.5	2.435
2.9	24.015	6.4	16.890	10.030	9.5	13.6	2.231
2.908	24.0	6.5	16.686	10.1	9.357	13.7	2.028
3.0	23.812	6.592	16.5	10.2	9.153	13.714	2.0
3.1	23.608	6.6	16.483	10.276	9.0	13.8	1.824
3.153	23.5	6.7	16.279	10.3	8.950	13.9	1.621
3.2	23.405	6.8	16.075	10.4	8.746	13.960	1.5
3.3	23.201	6.837	16.0	10.5	8.543	14.0	1.417
3.399	23.0	6.9	15.872	10.521	8.5	14.1	1.213
3.4	22.997	7.0	15.668	10.6	8.339	14.2	1.010
3.5	22.794	7.083	15.5	10.7	8.135	14.205	1.0
		7.1	15.864			14.3	0.8062
						14.4	0.6026
						14.45	0.5
						14.6	0.3990
						14.696	0

PART III

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15. INSULATION AND MOISTURE-PROOFING

Economic Factors

THE costs of insulating refrigerated rooms or pipes, vessels and other equipment are large enough to warrant careful consideration of type and quantity of material to use. **First cost** is important, but even more so is the **continuing cost** made up of operating expense, maintenance and repairs and depreciation.

In considering insulation of heated surfaces it is conventional to compare the cost of heat loss from bare surfaces with that from insulated ones, thus determining the sum economically available to offset the cost of insulation. Exactly the same approach is not possible for refrigeration as it is seldom practical to operate without any insulation. Similar basic analysis is appropriate however. As the amount or thickness of insulation increases its cost goes up but the cost of refrigeration decreases. The most economical thickness is therefore the point where the sum of these costs per year is least. **Fig. 1** illustrates this, where the annual costs of insulation and refrigeration are respectively curves N and M. (Mathematical form and derivation by McMillan shown in Appendix A.)

It is important to note that the thickness so determined may not be sufficient to withstand the conditions of service. Other factors may modify purely economic considerations.

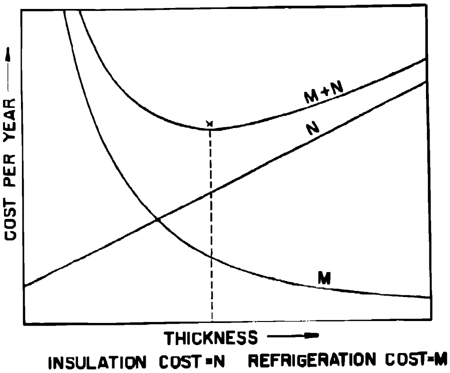


Fig. 1. X = **Most Economical Thickness**

Insulation Thickness in Practice

Extensive studies of all factors affecting the true economic thickness are warranted only for large permanent structures. The great majority of transient food storages held at 32 F to 40 F are insulated with 4 in. of sheet or block type insulation having a conductivity between 0.28 and 0.33 Btu per hr sq ft F per in. of thickness.

For other conditions, the thicknesses given in **Table 1** are in accord with general practice applying to the same type of insulation. Fill type insulations are generally used from ⅓ to ½ thicker. Manufacturers of different insulants have their own recommendations and will supply information on conductivity, moisture resistance and other properties.

Table 1

Storage temp F	Insulation thickness in inches	
	Northern U. S.	Southern U. S.
−45 to −15	9	10
−15 to 0	7	8
0 to 15	6	7
15 to 25	5	6
25 to 35	4	5
35 to 50	3	4
50 to 60	2	3

Unusual conditions of climate or use should modify the normal recommendations. Roofs should have 1 in. more insulation than walls in northern latitudes and 2 in. more in the South because of the greater equivalent temperature difference where the roof is exposed to the sun. Floors, except in first-floor freezers, may have 1 in. less than walls.

One rule of thumb method of estimating load or overall heat loss and hence refrigerating capacity required, that should not be used unless properly modified by size factors, is "so many Btu per 100 cubic feet of capacity." Heat entry through insulation depends on total surface area times rate, and a cube of 1,000,000 cu ft has one-tenth the proportionate surface of a cube containing 1,000 cu ft.

Types of Structure

Three possible types of structure present as many problems for consideration:

1. Experimental construction, used only through the life of the test.
2. Temporary construction, useful up to five years.
3. Permanent construction, for the life of the structure housing the insulation.

Experimental construction obviously calls for the lowest possible investment. Failures due to condensation and freezing are unimportant, so long as they do not interfere with the experiment. For this purpose many fill type insulations, felted blankets and other cheap blocks are available. Wood framing and either wood, compo-board, or light metal sheathing may be employed and packed with the fill material.

Temporary construction including insulation should not be undertaken, unless low cost factors are paramount. For more than a brief period it will be found more costly in the end.

For **permanent construction,** insulating materials used should have lasting properties equal to those of the structure itself, which should be of brick and steel or concrete. If wood is used it must be specially protected against rot, vermin, and other destructive agencies.

Thermal Factors

As stated in Chapter 5, "Fundamental Concepts of Heat Transfer," **conductivities** of all insulations depend upon the density and vary directly with mean temperature and with respect to moisture content. As here used, **mean temperature** of the material is the arithmetic mean of surrounding air temperature and the operating temperature of the room or piece of equipment under consideration. Rowley, Jordan and Lander have reported[1] that conductivities of typical sheet or block type insulants decrease about 12.6% per 100 deg decrease in mean temperature. Except for a few chemical process and ultra-low temperature applications, the maximum difference in mean temperatures encountered will not exceed 50 deg F, so that the conductivity of insulation will not vary more than about 6% from one type of job to another.

There will be more variation than this in any carload of commercial insulation and still more is likely as a result of differing degrees of skill and care in applicators. Consequently there is no need in most cases to take the time to compute variation in conductivity due to mean temperature. Values determined at a mean of 50 or 60 F will be within 3 or 4% of accuracy for almost any job.

Variation in conductivity due to **moisture content** cannot be dismissed so easily. While there has not been general agreement on the extent to which conductivity is increased as the moisture content increases, there is full agreement that wet insulation is useless and a structural hazard. It is also agreed that unless properly protected against the entry of moisture, usually in the form of water vapor, nearly all insulations will soon become saturated, and structural damage either by corrosion or the expansive action of freezing water will follow.

The entry of water vapor into insulation is primarily the result of difference in pressure due to temperature. The temperature gradient through an insulated wall is proportional to the thermal resistance of the layers or lamina considered, in relation to the total resistance of all in series. This is shown in **Fig. 2,** from which it is apparent also that the place to put the protection against moisture, usually called the **vapor barrier,** is at the warm side of the insulation, since most of the material between that point and the cold surface is below the dew point. Vapor which gets through the barrier may, if its passage is not seriously impeded, continue traveling toward the region of still lower temperature and pressure, and wind up as frost on the cooling surfaces. Whether it does so or col-

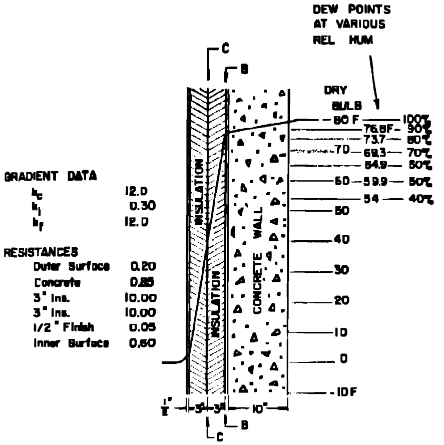


Fig. 2. Conventional Temperature Gradient. (Shows that vapor barrier at B-B near warm side is preferable to barrier at C-C closer to cold side)

lects in the insulation depends on the characteristics of the latter, the amount of vapor present and the presence or absence of more or less effective barriers. The only safe practice is to provide as good a seal or barrier on the warm side of the insulation as is practicable. It follows that the absence of vapor barriers or surfaces of low permeability between the warm side and the cold surface is desirable.

One type of installation that requires special treatment is an enclosure which is alternately warmer and cooler than its surroundings. If the temperature differences are large, and particularly if high humidities are involved, present state of knowledge indicates that light sheet metal surfacings with joints soldered or welded are required. In railroad car design there is a tendency toward assuming that because of vibratory strains, sealed surfaces will not remain tight, moisture will enter and must be allowed to escape.

All materials resist the passage of gases, yet all materials allow it to some degree. Moisture in the atmosphere performs as a gas, having the property of diffusing throughout the gases of the air, and through most materials at a fairly rapid rate. Moisture will diffuse and pass through barriers at a rate proportional to its vapor pressure and inversely proportional to the resistance of the barrier. **Vapor pressure** refers to the pressure of the moisture (or steam) only, independent of the air. The rate at which moisture will pass through some materials is shown on page 256.

Table 2. Vapor Pressure Values

Temperature, F	Water vapor pressure at saturation, lb per sq ft	Temperature, F	Water vapor pressure at saturation, lb per sq ft
−45	0.205	30	11.62
−40	0.268	35	14.40
−35	0.367	40	17.50
−30	0.496	45	21.25
−25	0.669	50	25.61
−20	0.892	55	30.82
−15	1.182	60	36.9
−10	1.562	65	43.9
− 5	2.05	70	52.3
0	2.67	75	61.8
5	3.46	80	73.0
10	4.45	85	85.7
15	5.72	90	100.7
20	7.26	95	117.2
25	9.22	100	136.7

From **Table 2** it is apparent that quite considerable pressures are involved in most refrigerated structures. The pressure differential between water vapor in the air at 80 F and 80% relative humidity and at 0 F and 85% relative humidity is slightly over 56 lb per sq ft.

Fig. 3, representing the same construction as in Fig. 2 but with two temperature gradients ((a) as in Fig. 2 for 6-in. insulation, with a conductivity of 0.27, and (b) for 6-in. insulation with a conductivity of 0.60), illustrates that within the conductivity range of the commercial insulants (0.27 to 0.45) the location of the vapor barrier need not change, even though there is a considerable change in the amount of heat flowing. However, if the structural part of the wall supplies enough of the total resistance to cause a temperature drop through it equal to or exceeding the assumed dew point depression (as in gradient b if relative humidity is 80%), then the vapor protection must be on its outside surface. For Fig. 3, the gradient figures and heat flow rates for both insulations

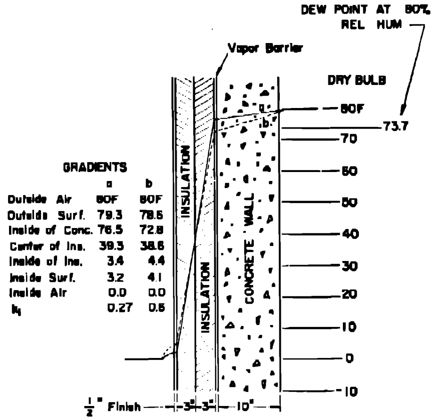


Fig. 3. Gradient Variation with Change in *k* of 120%

and for the wall uninsulated are:

	Resistances			*Temperature Drops*		
	a *Ins.* ($k = .27$)	*b* *Ins.* ($k = .60$)	*c* *Unins.*	*a*	*b*	*c*
Outside surface	.20	.20	.20	0.7 deg	1.4 deg	9.8 deg
Concrete wall 10/12	.85	.85	.85	2.8	5.8	41.2
6-in. insulation	22.2	10.00	—	74.3	68.4	—
½-in. cement finish	.05	.05	—	0.2	0.4	—
Inside surface	.60	.60	.60	2.0	4.0	29.0
Totals	23.9	11.7	1.65	80.0 deg	80.0 deg	80.0 deg
Heat flow $= \Delta t/R$ = Btu per hr sq ft	3.35	6.84	48.5	—	—	—

If only one point of the gradient needs to be determined, as for example the temperature of the cooler surface of the concrete wall in Fig. 3, a simple proportion will give it. If insulation of a conductivity of 0.45 is being considered ($R = 15.0$ and the resistance of surface and wall is 1.05), then $80 - (1.05/15.0 \times 80) = 74.4$ F, which is just above the dew point for the assumed conditions.

Surface Condensation

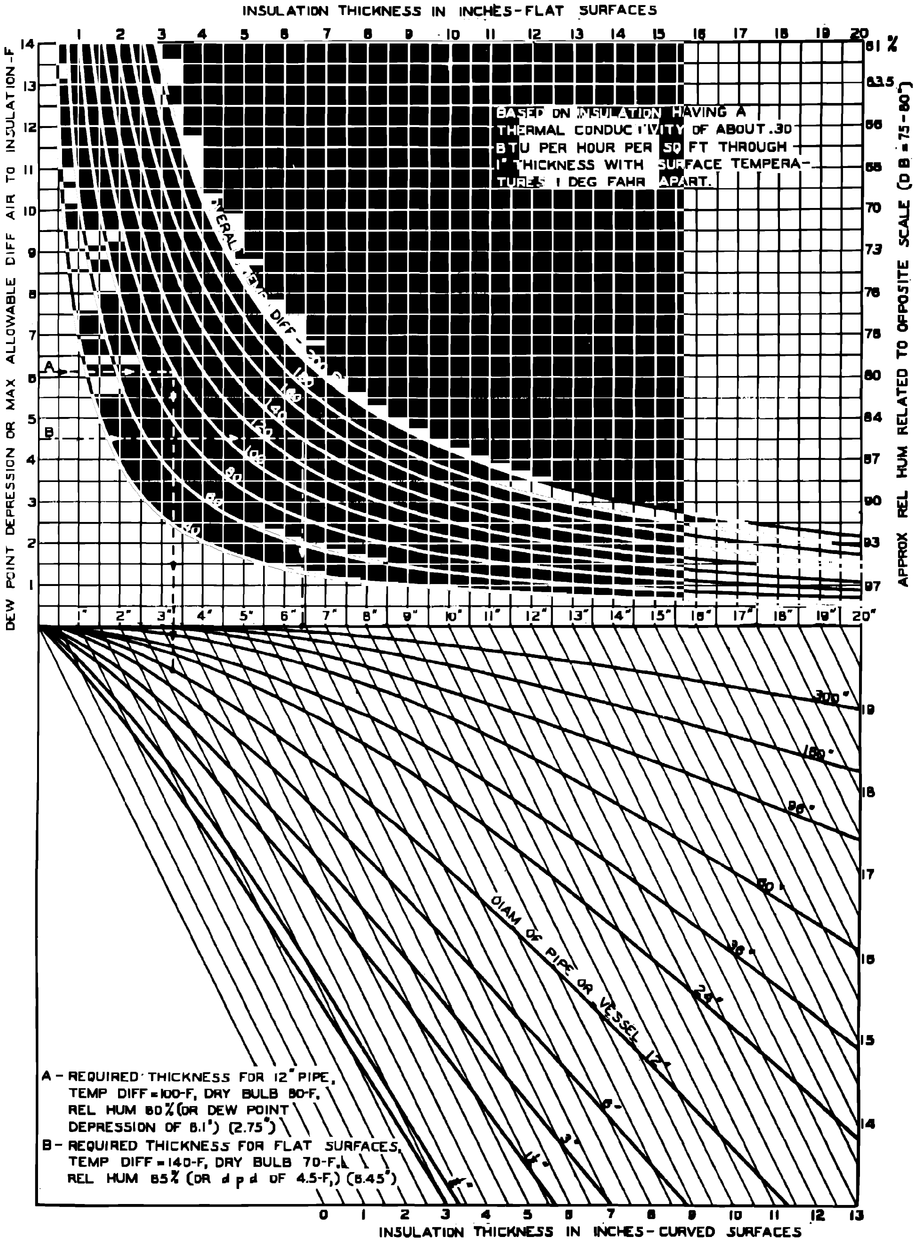
To prevent condensation on the outside surface of insulated equipment or rooms requires the use of a sufficient thickness of insulation to insure that the temperature drop from the surrounding air to the insulation surface is less than the dew point depression. For ordinary conditions this means about 1 in. of insulation for each 20 to 25 deg F of overall temperature difference.

The amount required varies considerably depending on the location. In humid climates more insulation is needed, as will be seen from Fig. 4. As an example, with the air temperature at 80 F, and 85% relative humidity, the dew-point depression is approximately 4.6 deg, while if the relative humidity is 60% the dew-point depression is about 14.5 deg. To get the minimum thickness of insulation for each condition, pick out the dew-point depression at left side of figure, run a line horizontally to the overall temperature difference line, say 80 F if refrigeration temperature is 0 F, and drop to bottom of chart. For flat surfaces and 4.6 deg dew-point depression the thickness is 3.5 in., but for 14.5 deg dew-point depression it is just under 1 in.

One thing that cannot be done is to insulate heavily enough to prevent condensation under any conditions. To stop condensation at 100% relative humidity would require an infinite thickness of insulation, since the heat flow would have to be nil.

Foundations and Freezer Floors

For storage temperatures of 25 F and higher, foundations and areas beneath floors present no different problems than in non-refrigerated buildings. When the sub-floor is on the ground it is particularly desirable to use an insulation that absorbs a minimum of moisture, or provide thorough waterproofing beneath it. If freezer space is contemplated on or below grade, consideration should be given to the possibility of freezing the ground and heaving the floor or column and wall footings. Ac-



BASED ON INSULATION HAVING A THERMAL CONDUCTIVITY OF ABOUT .30 BTU PER HOUR PER SQ FT THROUGH 1" THICKNESS WITH SURFACE TEMPERATURES 1 DEG FAHR APART.

A - REQUIRED THICKNESS FOR 12" PIPE, TEMP DIFF = 100-F, DRY BULB 80-F, REL HUM 80% (OR DEW POINT DEPRESSION OF 8.1°) (2.75")

B - REQUIRED THICKNESS FOR FLAT SURFACES, TEMP DIFF = 140-F, DRY BULB 70-F, REL HUM 85% (OR d p d OF 4.5-F,) (6.45")

Fig. 4. Insulation Thickness to Prevent Surface Condensation

cording to Cooling and Ward[2] the risk of this happening becomes considerable with storage temperatures below 15 F, of course being greater as the temperatures go lower. The most unfavorable soils are of the fine-grained type such as fine sands, silts and clays.

Positive **drainage** is essential and must be provided for by the use of gravel or cinder fills plus drain tile, if there is any reason to fear seepage of ground water. For low-temperature freezing or holding rooms a depth of dry fill of 6 ft beneath the concrete floor slab is not excessive, although the use of heavier insulation on the floor would be a way of reducing the depth of fill. What is needed is to balance the heat subtracted from the ground beneath the floor with the heat flowing into the cooled area from the surrounding earth, so that a temperature of 32 F or lower is not reached in the ground beneath the dry fill. If it is, and there is moisture present, layers of ice form and build up in thickness, mainly drawing water from the surrounding area by capillary action.

In extreme cases where freezer rooms have to be on the lowest floor and the ground water level may approach the sub-floor slab, the only way to be safe is to provide for supplying enough heat beneath the floor insulation to balance that flowing up into the cold room when the ground temperature is 33 to 35 F.

This can be done by making the sub-floor slab a form of panel heating. Non-corroding pipe or tubing in the concrete carrying heated oil is being used by one engineer in the Mid-West to meet such conditions and seems a simple, effective and reliable method. Other suggestions include electric strip heaters and air channels beneath the slab carrying heated air. The designer confronted by such a problem will find it helpful to consult Professor C. F. Kayan's paper on "Temperatures and Heat Flow for a Concrete Slab with Imbedded Pipes."[3]

One type of insulation that has to be considered separately is known as reflective insulation. There are many variations but all depend on surfaces having low emissivities or high reflective values for radiant heat. The principle is not new and was used to some extent in laboratory experiments before 1890. The Dewar flask which utilizes the reflective effect of silvered glass surfaces plus a vacuum between them was invented in 1892.

In present day practice reflective insulation may be metal foils, foil clad paper or metal sheets. Thicknesses range from 0.0003 in. to 0.006 in. or more. Methods of installation range from crumpling foil by hand and packing in the space to be filled to applying foil or sheets to wood or other spacers. Aluminum has been used more than any other metal but lead-tin coated steel sheets have also been used and Wilkes[3] mentions in addition platinum foil, nickel foil, and gold foil as suitable for high temperature experimental work.

The use of reflective surfaces to reduce transfer of heat by radiation requires that there be an air space facing the surface. If both surfaces of the space have low emissivities of the order of 0.05 to 0.10, the radiant transfer is greatly reduced and the main transfer is by conduction through the air and by convection.

By the use of multiple spaces a high resistance to heat flow is developed. The effectiveness of the construction depends to a certain extent on position and dimensions. In horizontal position with heat flow downward, as in the ceiling of a cold room, there is little convection, and the width of the spaces is important. Wilkes[4] shows that with a mean temperature of 75 F and surface emissivities of 0.05, five spaces give approximately the following conductances:

Width of space	*Conductance*
½ in.	0.10 Btu per hr sq ft F
¾ in.	0.07 Btu per hr sq ft F
1⅝ in.	0.03 Btu per hr sq ft F

In a vertical position the ¾ in. and 1⅝ in. spaces are of nearly equal value. Increasing the space beyond 1⅝ in. would have little effect while decreasing it below ¾ in. would increase the heat transfer.

Protection against moisture is required for reflective insulation as much as for any other. The value depends on retention of the low emissivity which would be greatly increased by corrosion or dirt. According to Taylor and Edwards:[5] "A piece of aluminum may have very high reflectivity for visible light and low reflectivity for infra-red radiation or it may have only fair reflectivity for light and be an excellent reflector of infra-red radiation, depending on the presence or absence of surface films."

Piping and Vessels

The insulation of piping, tanks and vessels differs in one important respect from insulating rooms and buildings. In the case of the latter, the applicator is nearly always building inwards, that is, from what will be the warm surface of the insulation toward what will be the cold side. His problem is primarily to make vapor-tight, or nearly so, the surface of the first layer of insulation itself. Exactly the reverse is true on piping and tanks, and it is the outer finish or protection that will determine to a large degree the permanence of the job.

Only two general types of insulation have been used to date with any success on piping and vessels:

1. The relatively **solid type** made commercially in the form of semi-cylindrical sections for pipes and as bevelled lags and blocks for large pipes, vessels and tanks;

2. The comparatively soft, springy **blanket type** typified by hair felt. Recently there has been some use of felted mineral wool blankets, applied in much the same manner as hair felt.

Engineers designing a job, or owners buying insulation, must make their own choice between the two, for undeniably there have been first class installations of both types giving long continued good service. It is equally undeniable that there is considerably more difficulty in making sure of a good protective seal against moisture over the softer type of material.

Fittings are insulated either by covering them with molded or milled jackets of the same material as the sections of pipe insulation, or by wrapping them with one of the blanket type materials. In either case it is necessary to take particular precautions that the sealing be good where there are projections of metal through the insulation, such as valve stems, hangers, etc.

Table 3 gives heat transmission values for **solid sectional type insulation** having a nominal thermal conductivity of the order of 0.30 Btu per hr per sq ft of surface per deg F temperature difference between surfaces. **Table 4** shows minimum **spacing** that must be allowed to permit proper application and sealing of pipe insulation.

The insulation of **water piping** required to prevent freezing when the pipe is exposed to low atmospheric temperatures is shown in **Table 5** for both types of insulation. It must be remembered that if there is no flow of water and the surrounding temperatures remain low, no amount of insulation will prevent freezing.

Insulation thicknesses to prevent **surface condensation** are shown in Fig. 4. To figure any special case it is necessary to know the dew-point depression in degrees below the expected ambient, the overall temperature difference and the commercial conductivity of the insulation it is proposed to use. The surface resistance value can be taken as 0.65, which gives satisfactory results and provides a slight margin of safety in most cases. For flat or large diameter surfaces the equation is

$$x = 0.65k\left(\frac{t_a - t_o}{t_a - t_s} - 1\right),$$

where x = Thickness of insulation

k = Conductivity of insulation

t_a = Air temperature (ambient)

t_o = Temperature of surface to be insulated, frequently called operating temperature

t_s = allowable temperature of outer surface of insulation.

For pipes or vessels of moderate and small diameter there is no simple equation as the equivalent thickness, $r_2 \log_e (r_2/r_1)$ must be used in place of x. In this expression r_2 is the radius of the outer surface of the insulation and r_1 is the radius of the surface to be insulated.

The effect of diameter is small for small temperature differences but becomes considerable as $t_a - t_0$ increases. **Fig. 7** illustrates this for one value of $t_a - t_s$.

Several applications of insulation to tanks and vessels are shown in **Figs. 5** and **6**. Recommended methods vary somewhat between manufacturers of different insulants but the examples shown are quite representative.

Application Specifications

Typical specifications are as follows:

The surface to be insulated shall be brushed

Table 3. Heat Transmission through Sectional Pipe Insulation

The rates of heat transmission given below are expressed in Btu per sq ft (and also per linear ft), per hr. per degree temperature difference between fluid in the pipe and air surrounding the pipe. In the case of screwed fittings, where the proportion of fittings to pipe is not unusually high, no great error will be introduced by figuring small fittings, such as ells and tees, as 1 linear ft of pipe; and larger fittings, valves, crosses, etc., as 1½ linear ft. Reference to a table of areas will be necessary in computing radiation equivalents for flanged fittings. Conductances in this table are based on an insulation conductivity of 0.30 Btu per hr sq ft deg F per in. of thickness.

Pipe size in.	Ice water thickness			Brine thickness			Special or heavy brine thickness		
	Btu per lin ft	Btu per sq ft pipe surface	Btu per sq ft outer surface	Btu per lin ft	Btu per sq ft pipe surface	Btu per sq ft outer surface	Btu per lin ft	Btu per sq ft pipe surface	Btu per sq ft outer surface
½	.110	.502	.105	.098	.446	.075	.087	.394	.051
¾	.119	.431	.101	.111	.405	.085	.094	.340	.051
1	.139	.403	.118	.124	.352	.086	.104	.294	.053
1¼	.155	.357	.118	.131	.300	.077	.113	.260	.054
1½	.174	.351	.133	.134	.270	.073	.118	.238	.053
2	.200	.322	.139	.151	.244	.077	.134	.214	.057
2½	.228	.303	.145	.170	.226	.081	.147	.197	.059
3	.269	.293	.158	.186	.202	.079	.162	.176	.059
3½	.295	.282	.161	.191	.183	.073	.176	.167	.061
4	.294	.248	.140	.209	.176	.076	.182	.154	.058
5	.349	.239	.148	.241	.165	.080	.202	.138	.057
6	.404	.233	.154	.259	.150	.076	.228	.130	.060
8	.455	.201	.139	.318	.140	.081	.263	.116	.059
10	.559	.198	.147	.383	.135	.086	.309	.110	.062
12	.648	.194	.150	.438	.131	.088	.364	.108	.067
14	.670	.182	.142	.487	.132	.093	.392	.107	.068
16	.750	.179	.143	.542	.130	.094	.434	.104	.069
18	.833	.176	.145	.604	.127	.096	.476	.102	.070
20	.919	.175	.146	.661	.126	.097	.520	.100	.071

Table 4. Minimum Spacing Requirements

Insulation thickness	Space required between parallel pipes in.	Space required between pipe and adjacent surfaces in.
Ice Water Thickness (1½ in. to 2 in.)		
Up to 6-in. pipe—screwed fittings	7	5
Larger than 6-in. pipe—screwed fittings	11	6
All pipe sizes—flanged fittings	12	8
Brine Thickness (2 in. to 3 in.)		
Up to 6-in. pipe—screwed fittings	9	7
Larger than 6-in. pipe—screwed fittings	15	9
All pipe sizes—flanged fittings	16	10
Heavy Brine Thickness (3 in. to 4 in.)		
Up to 3-in. pipe—screwed fittings	11	9
Larger than 3-in. pipe—screwed fittings	19	13
All pipe sizes—flanged fittings	20	14

free of all loose and foreign materials and shall be thoroughly dry. Rough surfaces shall be filled out with a coat of Portland cement plaster consisting of one part Portland cement and two and one half parts clean sharp, sand. When dry the surface shall be primed with a thin coat of asphalt primer. For each 10 ft of wall height, a horizontal support shall be bolted to the wall to carry weight of insulation above.

Table 5. Insulation to Prevent Freezing of Water in Pipes

No thickness of any type of insulation will prevent freezing if there is no flow of water and the surrounding temperature remains low long enough.

The two tables shown are based upon (A) use of sectional pipe insulation having a conductivity of approximately .31 Btu per hr sq ft F per in. of thickness, and (B) hair felt or other flexible blanket having a conductivity of about .27.

Figures are based on air at −20 F, or a temperature difference of 62 deg, with entering water at 42 F. The time shown to lower water to 32 F is much less than would be needed to freeze all the water in the pipe, but when freezing starts the danger point has been reached.

	A				B			
Steel pipe size, in.	Insulation thickness	Btu per hr F per lin ft	Hours to cool from 42 to 32 deg F—no flow	Lb water flow per hr per lin ft to prevent freezing	Insulation, no. of layers each nom 1 in. thick	Btu per hr F per lin ft	Hr to cool from 42 to 32 F—no flow	Lb water flow per hr per lin ft to prevent freezing
½	Ice Water	0.110	0.43	0.63	2	.0895	.417	.537
	Brine	0.098	0.52	0.56	3	.0747	.500	.448
	Heavy Brine	0.087	0.73	0.50	4	.0660	.565	.396
1	Ice Water	0.139	0.78	0.79	2	.1125	.825	.675
	Brine	0.124	0.90	0.71	3	.0911	1.02	.548
	Heavy Brine	0.104	1.20	0.59	4	.0798	1.16	.480
1½	Ice Water	0.174	1.27	0.98	2	.1400	1.40	.840
	Brine	0.134	1.72	0.76	3	.1126	1.74	.676
	Heavy Brine	0.118	2.04	0.67	4	.0972	2.02	.583
2	Ice Water	0.200	1.69	1.14	2	.1586	1.94	.952
	Brine	0.151	2.31	0.86	3	.1244	2.48	.747
	Heavy Brine	0.134	2.71	0.76	4	.1063	2.90	.638
3	Ice Water	0.269	2.73	1.53	2	.2062	3.25	1.237
	Brine	0.186	4.02	1.06	3	.1572	4.27	.943
	Heavy Brine	0.162	4.68	0.93	4	.1322	5.08	.793
4	Ice Water	0.294	4.08	1.68	2	.2450	4.55	1.470
	Brine	0.209	5.84	1.19	3	.1850	6.02	1.110
	Heavy Brine	0.182	6.77	1.04	4	.1548	7.20	.929
6	Ice Water	0.404	6.45	2.30	2	.3302	7.35	1.981
	Brine	0.259	10.01	1.48	3	.2434	9.88	1.460
	Heavy Brine	0.228	11.61	1.30	4	.1984	12.20	1.191
8	Ice Water	0.455	9.75	2.60	2	.4100	10.05	2.460
	Brine	0.318	14.00	1.82	3	.2960	13.90	1.776
	Heavy Brine	0.263	17.00	1.50	4	.2390	17.25	1.434
10	Ice Water	0.559	12.22	3.20	2	.4930	13.00	2.960
	Brine	0.383	17.95	2.20	3	.3536	18.10	2.122
	Heavy Brine	0.309	22.32	1.76	4	.2830	22.70	1.698
12	Ice Water	0.648	14.80	3.70	2	.5720	15.80	3.432
	Brine	0.438	22.12	2.50	3	.4090	22.20	2.454
	Heavy Brine	0.364	26.80	2.08	4	.3222	28.10	1.933

Insulation shall consist of — inches of ———— applied in — layers. Each sheet shall be dipped on back surface only in hot asphalt at a temperature of 350 to 400 F, and immediately pressed firmly into place. Adhesion shall be uniform over entire area.

The second layer is to be similarly applied to wall over the first layer, with all joints staggered in relation to each other. The sheets may be additionally secured to each other by wood skewers and with galvanized nails through caps to treated wood grounds bolted to the wall.

All joints must be tightly butted and all voids eliminated. Surfaces of the insulation shall be made smooth to receive finish.

Finish over the exposed surface of insulation shall be a Portland cement finish, applied in two coats. The first coat shall be thin and well scratched. The second coat shall be float finished and blocked off in squares of suitable size.

Appendix A

Derivation of McMillan's formula for economic thickness.

Let cost of refrigeration or heat entry equal M, and

$$M = \frac{a}{\frac{x}{k} + R}$$

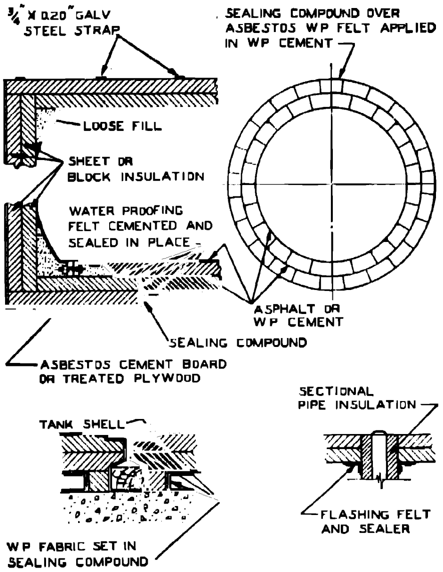

¾" × 0.20" GALV STEEL STRAP
SEALING COMPOUND OVER ASBESTOS WP FELT APPLIED IN WP CEMENT
LOOSE FILL
SHEET OR BLOCK INSULATION
WATER PROOFING FELT CEMENTED AND SEALED IN PLACE
ASPHALT OR WP CEMENT
SEALING COMPOUND
ASBESTOS CEMENT BOARD OR TREATED PLYWOOD
TANK SHELL
SECTIONAL PIPE INSULATION
FLASHING FELT AND SEALER
WP FABRIC SET IN SEALING COMPOUND

Fig. 5. Tank Insulation Details

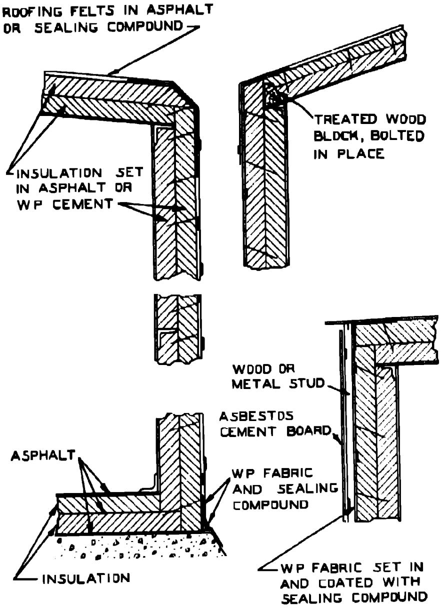

ROOFING FELTS IN ASPHALT OR SEALING COMPOUND
TREATED WOOD BLOCK, BOLTED IN PLACE
INSULATION SET IN ASPHALT OR WP CEMENT
WOOD OR METAL STUD
ASBESTOS CEMENT BOARD
ASPHALT
WP FABRIC AND SEALING COMPOUND
INSULATION
WP FABRIC SET IN AND COATED WITH SEALING COMPOUND

Fig. 6. Large Vertical Tank Insulation Details

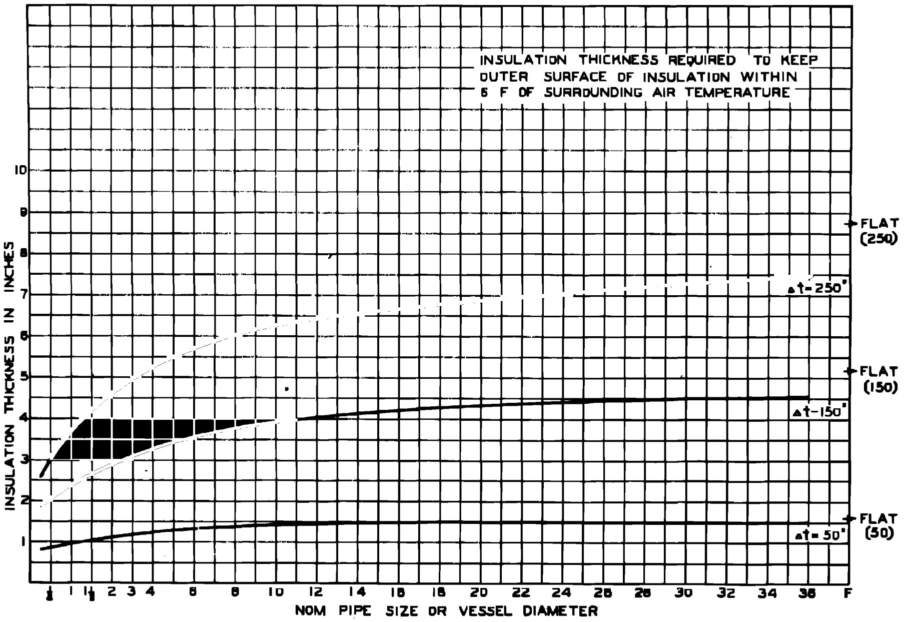


Fig. 7. Effect of Diameter on Insulation Thickness Required to Prevent Condensation

where

$$a = \frac{\text{hours operation per year} \times \Delta t \times \text{cost of refrig per ton day}}{288{,}000}$$

x = Thickness of insulation in inches
k = Conductivity of insulation in Btu/hr ft^{2}F/in.
R = Sum of all thermal resistances aside from insulation
b = Cost per year per inch of insulation per sq ft

Let the cost of insulation per square foot per year equal N, and $N = bx + C$. Let y = total cost per year $= M + N$

$$M + N = \frac{a}{\frac{x}{k} + R} + bx + C = \frac{ak}{x + Rk} + bx + C$$

Differentiating and equating the first derivative to zero:

$$\frac{dy}{dx} = \frac{-ak}{(x + Rk)^2} + b = 0$$

$$-ak + b(x + Rk)^2 = 0 \quad \text{or} \quad ak = b(x + Rk)^2$$

Dividing through by b and extracting the square root of both terms,

$$\left(\frac{ak}{b}\right)^{1/2} = x + Rk$$

Therefore for most economical thickness, $x = (ak/b)^{1/2} - Rk$.

References

1. Rowley, Jordan, and Lander; Thermal Conductivity of Insulating Materials at Low Temperatures; *Refrig. Eng.*, vol. 50, no. 6, p. 541; Dec. 1945.
2. Cooling, L. F. and W. H. Ward; Proceedings of the Inst. of Refrig., Great Britain, vol. XLI, 1944 5.
3. Wilkes, G. B.; Heat Insulation; John Wiley & Sons, Inc., New York, N Y.; 1940, p. 129.
4. Wilkes, G. B.; op. cit., p. 117.
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16. PIPE AND PIPING

THIS chapter deals with piping as a means of conveying fluid. Piping is also used as heat transfer surface, which use is treated in Chapter 5. Dimensions and construction details are the same for both uses, but other considerations differ.

The **design** of piping as a means of conveying fluid involves consideration of the following factors: Size, materials, strength, method of assembly, expansion and contraction, supports, protection against damage, control of flow, convenience, and restrictions imposed by codes, laws and regulations.

1. The **size** of pipe is determined principally by the permissible **frictional pressure drop.** How much drop is permissible depends on the cost of overcoming it. In refrigerant **suction lines,** whatever pressure drop occurs increases the power input per unit of refrigeration, and decreases capacity of apparatus, and it is generally eco-

Table 1. Dimensions of Welded Seamless Steel Pipe (ASA B36.1019–39)

Nominal pipe size, in.	Outside diameter, in.	Nominal Wall Thickness for Schedule Numbers, in.									
		10	20	30	40	60	80	100	120	140	160
1/8	0.405				**0.068**		**0.095**				
1/4	0.540				**0.088**		**0.119**				
3/8	0.675				**0.091**		**0.126**				
1/2	0.840				**0.109**		**0.147**				0.187
3/4	1.050				**0.113**		**0.154**				0.218
1	1.315				**0.133**		**0.179**				0.250
1 1/4	1.660				**0.140**		**0.191**				0.250
1 1/2	1.900				**0.145**		**0.200**				0.281
2	2.375				**0.154**		**0.218**				0.343
2 1/2	2.875				**0.203**		**0.276**				0.375
3	3.5				**0.216**		**0.300**				0.437
3 1/2	4.0				**0.226**		**0.318**				
4	4.5				**0.237**		**0.337**		0.437		0.531
5	5.563				**0.258**		**0.375**		0.500		0.625
6	6.625				**0.280**		**0.432**		0.562		0.718
8	8.625		0.250	**0.277**	**0.322**	0.406	**0.500**	0.593	0.718	0.812	0.906
10	10.75		0.250	**0.307**	**0.365**	**0.500**	0.593	0.718	0.843	1.000	1.125
12	12.75		0.250	**0.330**	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 OD	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 OD	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 OD	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 OD	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 OD	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 OD	30.0	0.312	0.500	0.625							

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses see appropriate material specifications.

Thicknesses shown in bold face type in Schedules-30 and 40 are identical with thicknesses for "standard weight" pipe in former lists; those in Schedules-60 and 80 are identical with thicknesses for "extra strong" pipe in former lists.

Table 2. Dimensions of Brass Pipe
(ASTM B43)

Nominal pipe size, in.	Outside diameter, in.	Wall thickness, in. Regular	Wall thickness, in. Extra strong
1/8	0.405	.062	.100
1/4	0.540	.082	.123
3/8	0.675	.090	.127
1/2	0.840	.107	.149
3/4	1.050	.114	.157
1	1.315	.126	.182
1¼	1.660	.146	.194
1½	1.900	.150	.203
2	2.375	.156	.221
2½	2.875	.187	.280
3	3.500	.219	.304
3½	4.000	.250	.321
4	4.500	.250	.341
5	5.563	.250	.375
6	6.625	.250	.437
8	8.625	.312	.500
10	10.750	.365	.500
12	12.750	.375	—

Table 3. Dimensions of Copper Tubing
(ASA H23.1-1941 or ASTM B88)

Nominal size, in.	Outside diameter, in.	Wall thickness, in. Type K	Wall thickness, in. Type L	Wall thickness, in. Type M
1/8	.250	.032	.025	.025
1/4	.375	.032	.030	.025
3/8	.500	.049	.035	.025
1/2	.625	.049	.040	.028
5/8	.750	.049	.042	.030
3/4	.875	.065	.045	.032
1	1.125	.065	.050	.035
1¼	1.375	.065	.055	.042
1½	1.625	.072	.060	.049
2	2.125	.083	.070	.058
2½	2.625	.095	.080	.065
3	3.125	.109	.090	.072
3½	3.625	.120	.100	.083
4	4.125	.134	.110	.095
5	5.125	.160	.125	.109
6	6.125	.192	.140	.122
8	8.125	.271	.200	.170
10	10.125	.338	.250	.212
12	12.125	.405	.280	.254

nomical to hold the pressure drop to a minimum. In **refrigerant liquid lines** a substantial pressure drop is permissible and involves no penalty, so long as the line will deliver the required amount of liquid to the evaporator. In **water** and **brine lines**, the cost of extra pumping head to overcome the **pressure drop** must be balanced against the cost of providing larger lines. In all cases where an economic balance is involved, the load factor must be taken into account.

Pressure drop depends on these factors: quantity, density and viscosity of the fluid, diameter, roughness and length of the pipe. Determination of pressure drop as a function of these factors is treated at length in Chapter 6. Length must include the equivalent length of valves, fittings, etc., as given in **Fig. 1**. Pipe and tubing come in certain "nominal" sizes which are given in **Tables 1, 2** and **3**.

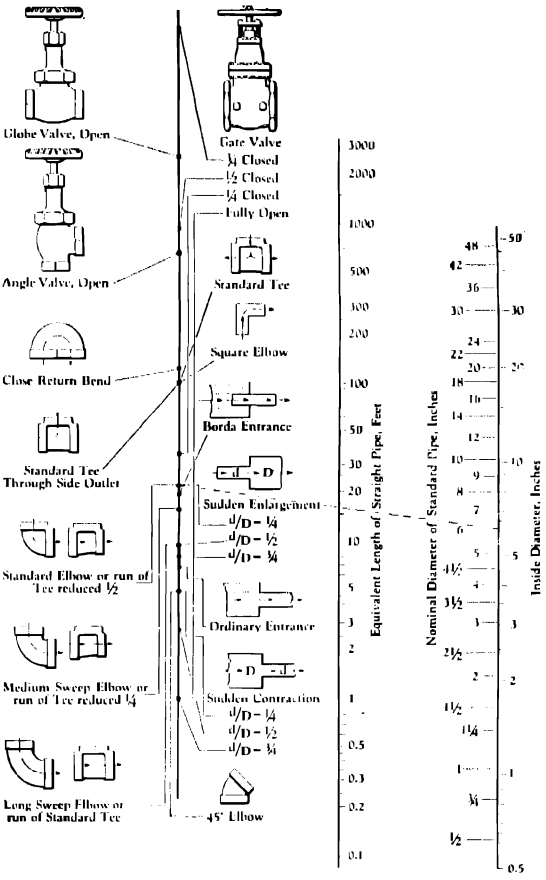
Tables 4a, 4b and **4c** give quantities (tons of refrigeration for refrigerants; gpm for water and brine) which will give a pressure drop of 1 psi in flowing through a 2-in. Schedule-40 steel pipe 100 ft long (including equivalent lengths). **Table 5** gives the multiplier, M_1, to apply to the flow in the 2-in. Schedule-40 steel pipe to obtain the same pressure drop in other nominal sizes of pipe and tubing. This pressure drop is rather high for refrigerant suction gases except for very short runs, and somewhat low for liquids except for very long runs. **Table 6** gives the multiplier, M_2, to apply to flow for obtaining other pressure drops.

For example, suppose it is desired to determine the quantity of flow of saturated ammonia vapor at 20 psig which will give a pressure drop of 1 psi in a 4-in. Schedule-40 steel pipe 200 ft long, with four elbows and one globe valve. The computation is as follows:

Total equivalent length in linear ft

Pipe	200
4 elbows at 11 ft (Fig. 1)	44
1 globe valve at 120 (Fig. 1)	120
Total	364

Permissible pressure drop per 100 lin ft $= 1/3.64 = .275$.



Crane Company

Fig. 1. Pressure Drop, Water Flow in Various Fittings

Example: 6-in. elbow is equivalent to 16½ ft of straight pipe

Multiplier for pressure drop (Table 6) $M_2 = .49$.

Multiplier for 4-in. Schedule-40 pipe (Table 5) $M_1 = 5.9$.

Tonnage in 2-in. Schedule-40 pipe for 1 psi, pressure drop per 100 lin ft (Table 4a) = 32.4 tons.

Tonnage in 4-in. Schedule-40 pipe for conditions given, $32.4 \times 0.49 \times 5.9 = 93.7$ tons.

A similar computation can be used to determine pipe size by solving for M_1 and referring to Table 5, although this involves trial and error in determining equivalent lengths, since these are different for the same type of fitting in different sizes.

The above formulation must be used with judgment and discretion, and if exact pressure drops are important, the method in Chapter 6 should be used, after determining approximate size by the method given. However, for ordinary sizing of pipe, there is such a great difference in pressure drop for a given flow between adjacent sizes that by using the above method, and picking the size which most nearly *exceeds* that indicated, the proper result will be obtained. In cases of doubt, oversizing is preferable to undersizing.

2. Materials commonly used in re-

Table 4a. Saturated Vapors. Flow Giving 1-psi Pressure Drop per 100 lin ft in 2-in. Schedule-40 ("Standard") Steel Pipe

Absolute pressure	Ammonia			Freon (F-12)			Freon (F-22)			Methyl Chloride			Sulfur Dioxide			Carbon Dioxide		
	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig
in. Hg	F			F			F			F			F			F		
5	−83.0	219	8.0	−84.7	626	2.1	−99.3	538	2.6									
10	−63.7	310	11.6	−63.1	886	3.2	−79.2	757	3.8				−26.2	593	6.9			
15	−51.3	380	14.4	−48.8	1,076	4.0	−66.1	928	4.7				−12.5	727	8.5			
20	−41.9	439	16.7	−38.0	1,238	4.8	−56.4	1,069	5.6				− 2.0	840	9.9			
25	−34.3	490	18.8	−29.0	1,383	5.4	−48.4	1,192	6.3	−18.5	838	10.3	6.6	945	11.1			
psi																		
15	−27.3	540	20.8	−20.8	1,525	6.1	−40.8	1,314	7.1	−10.0	927	11.4	14.4	1,040	12.3			
20	−16.6	623	24.2	− 8.2	1,753	7.2	−29.4	1,507	8.2	2.9	1,073	13.4	26.4	1,202	14.3			
25	− 8.0	696	27.2	2.2	1,953	8.3	−20.0	1,685	9.4	13.3	1,204	15.2	36.3	1,345	16.0			
30	− 0.6	762	30.0	11.1	2,135	9.2	−11.7	1,854	10.5	22.3	1,325	16.9	44.8	1,472	17.6			
35	5.9	822	32.5	18.9	2,305	10.1	− 4.9	1,994	11.4	29.7	1,430	18.3	52.2	1,595	19.1			
40	11.7	877	34.8	25.9	2,465	11.0	1.5	2,130	12.3	36.6	1,535	19.8	58.8	1,707	20.4			
50	21.7	980	39.2	38.3	2,760	12.6	12.4	2,400	14.1	48.4	1,720	22.4	70.4	1,915	22.8			
60	30.2	1,072	43.1	48.7	3,020	14.1	21.8	2,610	15.5	58.6	1,892	24.8	80.3	2,095	24.9			
80	44.4	1,239	50.0	66.3	3,495	16.8	37.4	3,030	18.4	76.1	2,220	29.4	96.9	2,445		−67.4	2,320	10.2
100	56.0	1,386	56.2	80.9	3,925	19.3	50.4	3,400	21.0	88.7	2,440		110.2	2,750		−57.6	2,610	11.6
120	66.0	1,520	62.0	93.4	4,320		61.5	3,750	23.4	101	2,655		121.5	3,040		−49.3	2,865	12.8
140	74.8	1,643	67.2	104.5	4,710		71.3	4,070	25.6	111	2,805		131.6	3,310		−42.0	3,100	14.0
160	82.6	1,760	72.2	114.5	5,070		80.1	4,380	27.8	120	2,910					−35.4	3,320	15.1
180	89.8	1,868		123.7	5,420		88.2	4,680		128	3,140					−29.4	3,530	16.1
200	96.3	1,970		132.1	5,770		95.6	4,940		140	3,325					−23.8	3,730	17.1
250	110.8	2,195					112.0	5,650								−11.6	4,190	19.3
300	123.2	2,390														− 1.1	4,630	21.5
400																16.6	5,470	25.2
500																31.3	6,240	28.2
600																43.9	7,030	30.9
700																55.0	7,830	32.8
800																65.0	8,670	33.1
900																74.2	9,670	33.1
1,000																82.6	11,100	30.0

"Lb per hr" is in 2-in. Schedule-40 steel pipe with 1-psi pressure drop per 100 linear feet.
"Tons refrig" is based on 86 F liquid up to expansion valve.
Tables 4a, 4b and 4c, except data on Freon-22, were prepared by Mr. M. B. Gelber, and are based on viscosity and density data and formulation appearing in 1942 *Refrigerating Data Book*. Freon-22 added from information received from Kinetic Chemicals, Inc.

Table 4b. Superheated Vapors. Flow Giving 1-psi Pressure Drop per 100 lin ft in 2-in. Schedule-40 ("Standard") Steel Pipe

Absolute pressure	Ammonia			Freon (F-12)			Freon (F-22)			Methyl Chloride			Sulfur Dioxide			Carbon Dioxide		
	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig
in. Hg	F			F			F			F			F			F		
25													8	938	13.1			
psi																		
15													28	1,021	14.1			
20													57	1,144	15.4			
25										25	1,203	17.8	81	1,254	16.6			
30				13	2,140	12.1				45	1,283	18.6	98	1,325	17.2			
35	7	822	38.6	22	2,310	12.7				65	1,367	19.5	115	1,413	18.0			
40	22	869	40.4	31	2,465	13.3				75	1,448	20.4	132	1,522	19.1			
50	47	965	44.0	46	2,730	14.2	16.4	2,370	18.1	100	1,612	22.1	158	1,667	20.4			
60	69	1,027	46.0	58	2,990	14.9	31.7	2,580	19.1	125	1,760	23.6	179	1,835	21.9			
80	105	1,153	50.1	78	3,450	16.0	56.7	2,940	20.8	155	1,990	25.6	215	2,030	23.3			
100	135	1,260	53.3	94	3,850	16.8	77.0	3,260	21.9	180	2,215	27.5	243	2,210	24.5			
120	160	1,354	56.1	109	4,220	17.4	94.0	3,550	22.8	200	2,430	29.2	265	2,405	26.0			
140	182	1,440	58.5	121	4,560	17.7	109.0	3,820	23.6	220	2,550	29.8	284	2,550	26.8			
160	201	1,520	60.6	133	4,890	17.9	122.3	4,070	24.2	235	2,680	30.5						
180	219	1,595	62.5	143	5,220	18.1	134.7	4,300	24.6	250	2,805	30.9						
200	235	1,660	64.0	152	5,520	18.2	145.7	4,530	25.0	270	2,940	29.8						
250	270	1,820	67.5				168.8	5,030	25.3									
300	300	1,955	70.1															
400																28	5,330	49.5
500																54	5,860	50.1
600																82	6,430	50.7
700																98	6,750	48.8
800																118	7,100	47.0
900																133	7,500	44.5
1,000																148	7,750	39.7

Temperature of compressed vapor based upon isentropic compression from saturated vapor at 5 F.
Tons of refrigeration based upon liquid at temperature corresponding to pressure to saturated vapor at 5 F.

Table 4c. Liquids. Flow Giving 1-psi Pressure Drop per 100 lin ft in 2-in. Schedule-40 ("Standard") Steel Pipe

Fluid	Temp, F	Density, lb/cu ft	lb/hr	Flow, gpm	Tons†
Water	32	62.4	15,900	31.8	
	100	62.0	17,100	34.5	
	212	59.8	17,800	37.2	
Water, average pipe*				25.0	
Sodium brine — 10% NaCl	25	67.2	16,200	31.2	
— 15% NaCl	15	69.9	16,100	28.8	
	25	69.8	16,300	29.2	
— 20% NaCl	5	72.7	15,500	26.6	
	15	72.5	15,900	27.4	
	25	72.2	16,350	28.3	
Calcium brine — 10% $CaCl_2$	30	68.2	16,350	30.0	
— 20% $CaCl_2$	0	74.7	15,200	25.4	
	15	74.5	15,800	26.5	
	30	74.3	16,500	27.7	
— 30% $CaCl_2$	−15	81.9	13,500	20.6	
	0	81.7	14,600	22.3	
	15	81.5	15,400	23.6	
	30	81.3	16,100	24.7	
Ammonia	86	37.2	14,100		556.
Carbon dioxide	86	37.4	17,900		82.8
Freon (F-12)	86	80.6	21,000		89.3
Freon-22 (F-22)	86	73.4	2,000		103.9
Methyl chloride	86	56.3	17,400		218.
Sulfur dioxide	86	84.4	21,300		251.

* Water pipe is subject to scale, corrosion and tuberculation. This value gives a close approximation to the Williams and Hazen tables, using $C = 100$.

† Tons refrigeration based on 5 F evaporation, saturated gas, and 86 F liquid.

frigerant, brine and water piping are steel, wrought iron, malleable iron, cast iron, copper, brass, and bronze, as furnished under the specifications listed in **Table 7**, which is taken from the ASA B31 Piping Code. Copper and brass must not be used with ammonia. Approval of use of any of the materials is predicated on refrigerants which are free from moisture and air, and on brine or water in which the *p*H value is properly controlled or in which proper **corrosion inhibitors** are used.

Other materials are not barred by the code, providing they are of suitable strength and chemically are suitable for use with the fluid conveyed. The code bars the use of magnesium base alloys with any Freon refrigerant; also the use of zinc, aluminum, die-castings and magnesium alloys with methyl chloride.

For temperatures below 0 F, the code provides that materials shall have a Charpy **impact value** at the operating temperature of not less than 15 ft-lb using a keyhole notch, but permits copper, brass and bronze without restrictions, and permits ferrous materials not meeting this requirement down to −100 F, provided the design pressure is increased 2% for each degree below 0 F. This permits the ordinary lines of cast iron refrigerant fittings down to −50 F without penalty. As to materials suitable for low temperatures, the code has the following to say:

"There is a large and constantly growing list of alloy compositions which meet

Table 5. Values of M_1 Multipliers to be Applied to Flow in 2-in. Schedule-40 Steel Pipe to Find Flow Giving Same Pressure Drop

Nominal size, in.	Steel or wrought iron Schedule-40, Standard	Schedule-80, Extra Strong	Brass Regular	Extra Strong	Copper tubing Type K	Type L	Type M
$\frac{1}{8}$	.0041	.0022	.0052	.0022	.00175	.0021	.0021
$\frac{1}{4}$	.0090	.0055	.0115	.0059	.0068	.0071	.0077
$\frac{3}{8}$	.021	.0138	.025	.0157	.0138	.0166	.0188
$\frac{1}{2}$	.039	.028	.047	.032	.029	.032	.036
$\frac{5}{8}$	—	—	—	—	.052	.055	.060
$\frac{3}{4}$	.086	.064	.098	.073	.075	.086	.094
1	.164	.127	.193	.144	.162	.176	.190
$1\frac{1}{4}$	.33	.27	.38	.31	.29	.30	.32
$1\frac{1}{2}$	.51	.42	.57	.48	.47	.49	.50
2	1.00	.84	1.13	.95	.99	1.02	1.04
$2\frac{1}{2}$	1.65	1.37	1.90	1.55	1.77	1.83	1.89
3	2.8	2.4	3.3	2.8	2.8	2.9	3.0
$3\frac{1}{2}$	4.2	3.6	4.6	4.2	4.2	4.4	4.5
4	5.9	5.1	6.6	5.8	6.0	6.2	6.3
5	10.7	9.5	12.1	10.6	10.6	11.0	11.2
6	17.5	15.3	19.9	16.9	16.8	17.6	17.9
8	36	32	40	35	35	36	37
10	67	58	71	66	61	64	66
12	108	93	114	—	97	103	104

these requirements, and it has not been deemed advisable to infer any limitation by including some few of them in Table 7. Depending on composition and heat treatment, some 'carbon steels' meet the requirements for a considerable range of temperature. In general, the 3 to 4 per cent nickel steels are acceptable for quite low temperatures, and the 18 per cent chrome-8 per cent nickel stainless steels meet requirements to very low temperatures. Various other classes of alloys are also acceptable. Apparently minor variations in composition and heat treatment make large differences in toughness at low temperatures."

A summary of available data is contained in "Properties of Metals at Subatmospheric Temperatures" issued in 1941 by the American Society of Testing Materials.

3. The **strength** of pipe, that is, the thickness required in a given material for a given pressure, is fixed by the Piping Code as follows:

$$t_m = \frac{PD}{2S+0.8P} + C$$

where

t_m = minimum pipe wall thickness in inches,

P = maximum internal service pressure in pounds per square inch gage (plus allowance for brittleness at low temperatures and water hammer allowance for cast iron pipe),

D = outside diameter of pipe in inches,

Table 6. Values of M_2 Multipliers for Flow to Give Other Pressure Drops

Pressure drop, psi	.0	.2	.4	.6	.8
0.	.00	.42	.61	.77	.89
1.	1.00	1.10	1.20	1.29	1.37
2.	1.45	1.52	1.60	1.67	1.73
3.	1.80	1.86	1.92	1.98	2.04
4.	2.10	2.15	2.21	2.26	2.31
5.	2.36	2.41	2.46	2.51	2.56

Table 7. List of Material Specifications

Material	Specification
Bolts	
Alloy steel	ASTM A96
Commercial steel (bar stock)	ASTM A107
Wrought iron (bar stock)	ASTM A84
Fittings, valves and flanges	
Cast iron flanges, flanged fittings and valves	ASTM A126
Cast iron bell and spigot fittings, and valves	AWWA
Malleable iron	ASTM A197
Steel (cast carbon)	ASTM A27 or A95
Steel (forged)	ASTM A105 or A181
Brass castings	ASTM B62
Bronze castings	ASTM B61
Pipe and tubing	
Steel, seamless or welded, for coiling or bending	ASTM A53
Steel, seamless or welded, for ordinary uses	ASTM A120
Steel, seamless or welded	API 5L
Steel, seamless or welded	ASTM A106
Steel, seamless	ASTM A83 or A179 or A192
Steel, seamless, medium carbon	ASTM A210
Steel, seamless, alloy	ASTM A213
Steel, electric resistance welded	ASTM A135 or A178 or A214 or A226
Steel, electric resistance welded, alloy	ASTM A249
Wrought iron, welded	ASTM A72
Wrought iron, welded	API 5L
Cast iron (pit cast)	ASA A21.2 (ASTM A44)
Cast iron (centrifugally cast or cast horizontally in green sand molds)	FSEC WWP-P-421
Brass (seamless)	ASTM B43
Copper pipe (seamless)	ASTM B42
Copper tubing	ASTM B88

S = allowable stress in material due to internal pressure in pounds per square inch, and

C = allowance for threading, mechanical strength and for corrosion, in inches, as follows:

Cast iron pipe centrifugally cast or cast horizontally in green sand molds, 0.14

Cast iron pipe, pit-cast, 0.18

Threaded steel, wrought iron or non-ferrous pipe, ⅜-in. and smaller, 0.05; for ½-in. and larger, use depth of thread.

Grooved steel, wrought iron or non-ferrous pipe, use depth of groove.

Plain end steel or wrought iron pipe, 1-in. size and smaller, 0.05; for sizes larger than 1 in., 0.065.

Plain end non-ferrous pipe or tube, zero.

Values of S to be used for the materials listed in Table 7 are given in **Table 8**. For other materials, the Piping Code gives rules for figuring S values. Values of service pressures to be used in the formula are two-thirds of the test pressures given in **Table 9,** but in no case are values of P less than 100 to be used.

The thicknesses of ferrous fittings and valves are covered by the various ASA B16 standards for various services, except that refrigerant fittings and valves have not been standardized, and "manufacturers standards of long standing" are recognized by the Code. However, ASA B16, 16 covering refrigerant fitting and flanges, was promulgated in 1948 and will be followed by all manufacturers after a transition period. There are ASA A40 standards on non-ferrous soldered and flared compression fittings.

There are also certain specific requirements set forth in the Piping Code and in the ASA B9 Safety Code for Mechanical Refrigeration. Chief of these rules are:

Refrigerant working pressure shall not exceed 250 psi in Schedule-40 ferrous pipe and butt welded pipe shall not exceed 2 in. in size.

Refrigerant liquid ferrous pipe, 1½-in. and smaller, shall be of Schedule-80.

Refrigerant tubing shall not be lighter than Type L for field assembly, or Type M for shop assembly.

Soft annealed copper tubing erected on the

Table 8. Allowable *S* Values to Be Used in Formula to Determine Pipe Thickness

Material	Specification	Value of *S*, psi
Steel pipe, seamless, Grade A	ASTM A53 or ASTM A120 or ASTM A106 or ASTM A83 or ASTM A179 or ASTM A192 or API 5L	12,000
Steel pipe, seamless, Grade B	ASTM A53 or ASTM A106 or ASTM A210	15,000
Steel, seamless alloy, grades T18, T19	ASTM A213	18,750
Steel, electric resistance welded, alloy grades T18, T19	ASTM A249	15,900
Steel pipe, lap welded	ASTM A53 or ASTM A106 or ASTM A120 or API 5L	9,000
Steel pipe, butt welded	ASTM A53 or ASTM A120	6,800
Steel pipe, electric resistance welded	ASTM A135 or ASTM A178 or ASTM A214 or ASTM A226	10,200
Wrought iron, lap welded	ASTM A72 or API 5L	8,000
Wrought iron, butt welded	ASTM A72	6,000
Cast iron pipe, pit cast	AWWA	4,000
Cast iron pipe, centrifugally cast or cast horizontally in green sand molds	FSEC WW-P-421	6,000
Brass pipe, seamless red brass	ASTM B43	4,700
Copper pipe, seamless	ASTM B42	4,000
Copper tubing, seamless	ASTM B88	4,000

premises shall not be used in sizes exceeding ¾ in.

Pipe thinner than Schedule-40 ferrous, or regular brass, shall not be threaded.

For refrigerants, cast iron shall conform to ASTM A126, Class B higher strength grey iron with not less than 31,000 psi tensile strength.

4. The **method of assembly** of piping refers to the type of joints, which may be screwed, flanged, welded, brazed, soldered, or in small tubing by flared compression fittings.

a. Screwed joints are limited by the Piping Code to not over 3 in. nominal size for refrigerant pressures 250 psi or less, 1¼ in. for refrigerant pressures above 250 psi and not over 4 in. for brine. Screwed-on flanges are not permitted in sizes larger than the above. Bushings are not permitted for reductions of a single pipe size. For screwed brass or copper refrigerant pipe, extra heavy fittings are required. **Table 10** gives dimensions of 125 and 250 psi screwed cast iron fittings up to 4 in. in size.

b. Flanged joints are used for sizes greater than the above, and in many cases for smaller sizes. There are various ASA B16 standards for flanged fittings for various services. The most commonly used are the 125 and 250 psi cast iron fittings,

Table 9. Test Pressures

Refrigerant	Chemical formula	Minimum test pressure, psi High side	 Low side
Ammonia	NH_3	300	150
Butane	C_4H_{10}	90	50
Carbon dioxide	CO_2	1,500	1,000
Dichlorodifluoromethane (Freon-12)	CCl_2F_2	235	145
Dichlorotetrafluoroethane (Freon-114)	$C_2Cl_2F_4$	80	50
Dichloromethane (Carrene No. 1) (Methylene chloride)	CH_2Cl_2	30	30
Dichloromonofluoromethane (Freon-21)	$CHCl_2F$	70	50
Dichloroethylene	$C_2H_2Cl_2$	30	30
Ethane	C_2H_6	1,100	600
Ethyl chloride	C_2H_5Cl	60	50
Isobutane	$(CH_3)_3CH$	130	75
Methyl chloride	CH_3Cl	215	125
Methyl formate	$HCOOCH_3$	50	50
Propane	C_3H_8	325	210
Sulfur dioxide	SO_2	170	95
Trichloromonofluoromethane (Freon-11)	CCl_3F	50	30

Note: For refrigerants not listed in table, high-side test pressure is saturation pressure at 150 F, low-side test pressure saturation pressure at 115 F, in no case less than 30 psi.

dimensions of which are given in **Fig. 2** and **Table 11.** The 150 and 300 psi steel standards conform to the same dimensions respectively.

Refrigerant flanged fittings and flanges were standardized in 1948. Standard dimensions are given in Tables 12 and 12A. These may be expected, during the next few years, to supersede manufacturers' standards which differ somewhat among

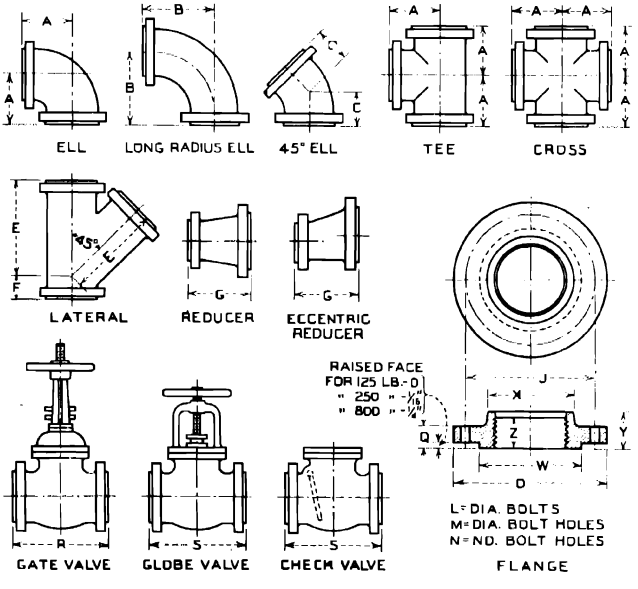
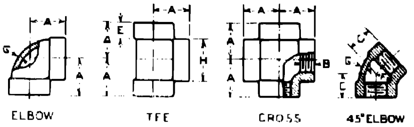


Fig. 2. Dimensions of Flanged Cast Iron Fittings, Flanges and Valves (see Table 11)

Table 10. Dimensions of Screwed Cast Iron Fittings, Straight Sizes

ASA B16d-1941



		A	C	B	E	F		G	H
	Nominal pipe size, in.	**Center to end, elbows, tees and crosses**	**Center to end, 45-deg elbows**	**Length of thread, min in.**	**Width of band, min in.**	**Inside diameter of fitting, in.**		**Metal thickness,* in.**	**Outside diameter of band, min in.**
						Min	**Max**		
125-psi	¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
	⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
	½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
	¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
	1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
	1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
	1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
	2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
	2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
	3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
	3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
	4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
250-psi	¼	0.94	0.81	0.43	0.49	0.540	0.584	0.18	1.17
	⅜	1.06	0.88	0.47	0.55	0.675	0.719	0.18	1.36
	½	1.25	1.00	0.57	0.60	0.840	0.897	0.20	1.59
	¾	1.44	1.13	0.64	0.68	1.050	1.107	0.23	1.88
	1	1.63	1.31	0.75	0.76	1.315	1.385	0.28	2.24
	1¼	1.94	1.50	0.84	0.88	1.660	1.730	0.33	2.73
	1½	2.13	1.69	0.87	0.97	1.900	1.970	0.35	3.07
	2	2.50	2.00	1.00	1.12	2.375	2.445	0.39	3.74
	2½	2.94	2.25	1.17	1.30	2.875	2.975	0.43	4.60
	3	3.38	2.50	1.23	1.40	3.500	3.600	0.48	5.36
	3½	3.75	2.63	1.28	1.49	4.000	4.100	0.52	5.98
	4	4.13	2.81	1.33	1.57	4.500	4.600	0.56	6.61

Reducing sizes for 125 psi vary slightly and are given in the B 16d Standard.
Reducing sizes for 250 psi are not covered by the Standard.
* Minimum at any point, 90% of value given.

themselves but conform more or less closely to the standard. The Piping Code provides that facing of refrigerant flanges shall be of the retained gasket type (male and female, tongue-and-groove, or ring) and all the manufacturers' standards are of the tongue-and-groove type. It is necessary to match flanges to be sure they come from the same manufacturer and to be sure that one of the pair is tongued and one grooved, and this frequently leads to annoying delays and extra joints in lines unless it is watched carefully. It also requires the carrying of larger inventories in plants or shops using equipment from more than one manufacturer.

c. **Welding** is practically universally accepted as the standard method of assembly of ferrous piping systems today, except valves are flanged and flanged unions are used where disassembly is frequent. Rules for qualification of welding procedures and operators are given at length in the Piping Code. This is a complex subject in itself and no brief description here would be other than misleading.

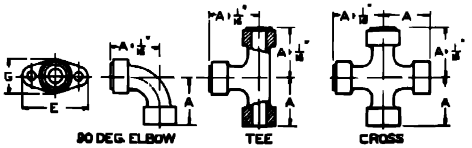
Table 11. Dimensions of Flanged Cast Iron Fittings and Valves (see Fig. 2)

	Nominal pipe size, in.	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
125-psi Standard ASA B16a-1939	A	3½	3¾	4	4½	5	5½	6	6½	7½	8	9	11	12
	B	5	5½	6	6½	7	7¾	8½	9	10¼	11½	14	16½	19
	C	1¾	2	2½	2¼	3	3	3½	4	4½	5	5½	6½	7½
	E	5¾	6¼	7	8	9½	10	11½	12	13½	14½	17½	20½	24½
	F	1¾	1¾	2	2½	2½	3	3	3	3½	3½	4½	5	5½
	G	—	—	—	5	5½	6	6½	7	8	9	11	12	14
	J	3⅛	3½	3⅞	4¾	5½	6	7	7½	8½	9½	11¾	14¼	17
	L	½	½	½	⅝	⅝	⅝	⅝	⅝	¾	¾	¾	⅞	⅞
	M	⅝	⅝	⅝	¾	¾	¾	¾	¾	⅞	⅞	⅞	1	1
	N	4	4	4	4	4	4	8	8	8	8	8	12	12
	O	4¼	4⅝	5	6	7	7½	8½	9	10	11	13½	16	19
	Q	7/16	½	9/16	⅝	11/16	¾	13/16	15/16	15/16	1	1⅛	1 3/16	1¼
	R*	—	—	—	7	7½	8	8½	9	10	10½	11½	13	14
	S*	—	—	—	8	8½	9½	10½	11½	13	14	19½	—	—
	X	1 15/16	2 5/16	2 9/16	3 1/16	3 9/16	4¼	4 13/16	5 5/16	6 7/16	7 9/16	9 11/16	11 15/16	14 1/16
	Y	11/16	13/16	⅞	1	1⅛	1 3/16	1¼	1 5/16	1 7/16	1 9/16	1¾	1 15/16	2 3/16
	Z	Same as Y												
250-psi Standard ASA B16b-1944	A	—	—	—	5	5½	6	6½	7	8	8½	10	11½	13
	B	—	—	—	6½	7	7¾	8½	9	10¼	11½	14	16½	19
	C	—	—	—	3	3½	3½	4	4½	5	5½	6	7	8
	E	6½	7¼	8½	9	10½	11	12½	13½	15	17½	20½	24	27½
	F	2	2¼	2½	2½	2½	3	3	3	3½	4	5	5½	6
	G	—	—	—	5	5½	6	6½	7	8	9	11	12	14
	J	3½	3⅞	4½	5	5⅞	6⅝	7¼	7⅞	9¼	10⅝	13	15¼	17¾
	L	⅝	⅝	¾	⅝	¾	¾	¾	¾	¾	¾	⅞	1	1⅛
	M	¾	¾	⅞	¾	⅞	⅞	⅞	⅞	⅞	⅞	1	1⅛	1¼
	N	4	4	4	8	8	8	8	8	8	12	12	16	16
	O	4⅞	5¼	6⅛	6½	7½	8¼	9	10	11	12½	15	17½	20½
	Q	11/16	¾	13/16	⅞	1	1⅛	1 3/16	1¼	1⅜	1 7/16	1⅝	1⅞	2
	R*	—	—	—	8½	9½	11⅛	11⅞	12	15	15⅞	16½	18	19¾
	S*	—	—	—	10½	11½	12½	13¼	14	15¾	17½	21	24½†	28†
	W	2 11/16	3 1/16	3 9/16	4 3/16	4 15/16	5 11/16	6 5/16	6 15/16	8 5/16	9 11/16	11 15/16	14 1/16	16 7/16
	X	2 1/16	2½	2¾	3 5/16	3 15/16	4⅝	5¼	5¾	7	8⅛	10¼	12⅝	14¾
	Y	⅞	1	1⅛	1¼	1 7/16	1 9/16	1⅝	1¾	1⅞	1 15/16	2 3/16	2⅜	2 9/16
	Z	0.68	0.76	0.87	1.00	1.14	1.20	1.25	1.30	1.41	1.51	1.71	1.92	2.12
800-psi Standard ASA B16b1-1931	A	—	—	—	5¾	6½	7	7½	8½	10	11	13	15½	16½
	C	—	—	—	4¼	4½	5	5½	6	7	7½	8½	9½	10
	E	—	—	—	10¼	11½	12¾	14	16½	19½	21	24½	29½	31½
	F	—	—	—	3½	3½	4	4½	4½	6	6½	7	8	8½
	G	—	—	—	6	6¾	7¼	7¾	8¾	10¼	11¼	13¼	15¾	16¾
	J	—	—	—	5	5⅞	6⅝	7¼	8½	10½	11½	13¾	17	19¼
	L	—	—	—	⅝	¾	¾	⅞	⅞	1	1	1⅛	1¼	1¼
	M	—	—	—	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1⅜	1⅜
	N	—	—	—	8	8	8	8	8	8	12	12	16	20
	O	—	—	—	6½	7½	8¼	9	10¾	13	14	16½	20	22
	Q	—	—	—	1½	1⅝	1¾	1⅞	2⅛	2⅜	2½	2¾	3⅛	3¼
	R*	—	—	—	11½	13	14	—	17	—	22	26	31	33
	S*	This dimension has not been standardized, except for check valves same as R												
	W	—	—	—	3⅝	4⅛	5	5½	6 3/16	7 5/16	8½	10⅝	12¾	15

* ASA B16.10.

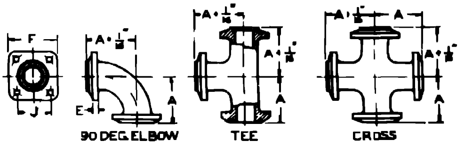
† Check valves only.

Table 12. ASA B16.16 Standard Cast Iron Refrigerant Fittings



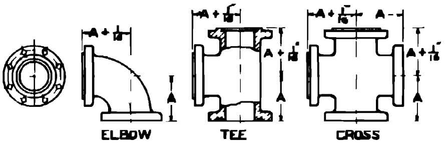
Oval Flanged Fittings

Nominal pipe size, in.	Inside diameter of fitting	Metal thickness of fitting, min	Long diameter of flange E	Short diameter of flange G	Thickness of flange, min B	Thickness† of flange at bolt circle D	Center* to female flange edge, elbow tee and cross A	Diameter of bolt circle	Number of bolts	Size of bolts
1/4	3/8	1/4	3 7/16	1 13/16	13/16	1/2	2 3/8	2 1/4	2	1/2
3/8	1/2	1/4	3 11/16	1 13/16	15/16	9/16	2 1/2	2 1/2	2	1/2
1/2	5/8	1/4	3 7/8	1 13/16	1	5/8	2 3/4	2 3/4	2	1/2
3/4	13/16	1/4	4 1/8	2 1/8	1 1/8	11/16	3 1/4	3	2	1/2



Square Flanged Fittings

Nominal pipe size, in.	Inside diameter of fitting	Metal thickness of fitting, min	Length and width of flange F	Thickness† of flange, min B	Thickness of flanges at edge E	Center* to flange edge, elbow tee and cross A	Bolt spacing J	Number of bolts	Size of bolts
1	1 1/16	1/4	3 9/16	15/16	9/16	3 3/8	2 5/16	4	1/2
1 1/4	1 3/8	5/16	3 7/8	1 1/16	5/8	3 7/8	2 3/8	4	1/2
1 1/2	1 9/16	5/16	4 1/8	1 1/8	11/16	4 1/4	2 13/16	4	1/2
2	2 1/16	3/8	5	1 1/4	13/16	4 3/4	3 7/16	4	5/8
2 1/2	2 1/2	3/8	5 1/2	1 3/8	7/8	5 3/8	3 7/8	4	5/8
3	3	7/16	6 1/4	1 1/2	15/16	5 7/8	4 5/16	4	3/4
3 1/2	3 1/2	7/16	6 1/2	1 5/8	1	6 3/8	4 3/4	4	3/4
4	4	7/16	7 1/4	1 3/4	1 1/8	6 7/8	5 3/8	4	7/8



Round Flanged Fittings

Nominal pipe size	Inside diameter of fitting	Wall thickness of fitting	Outside diameter of flange	Thickness† of flange,[1] [2] min		Center* to groove flange edge elbow, tee and cross A	Diameter of bolt circle	Number of bolts	Size of bolts
				Tongue	Groove				
5	5	11/16	11	1 5/8	1 9/16	8 3/16	9 1/4	8	3/4
6	6	3/4	12 1/2	1 11/16	1 5/8	8 11/16	10 5/8	12	3/4
8	8	13/16	15	1 7/8	1 13/16	10 3/16	13	12	7/8
10	10	15/16	17 1/2	2 1/8	2 1/16	11 11/16	15 1/4	16	1
12	12	1	20 1/2	2 1/4	2 3/16	13 3/16	17 3/4	16	1 1/8

* The 5/16-in tongue of oval and square flanges, and the 1/4" tongue of round flanges, is included in the "thickness of flange minimum," and all but 1/16 in is included in the "center to flange edge" dimensions

† See Table 12a

Welded joints are generally either of the butt-welding or socket-welding type. Dimensions of welding fittings have recently been standardized, or standards are in proposed form, and dimensions are given in **Tables 13, 14** and **15.** Branch connections require suitable reinforcing. Butt-welding is commonly used for 2-in. nominal size and larger, socket-welding for $1\frac{1}{2}$-in. and smaller.

d. Soldering for non-ferrous materials corresponds in function, although not in method, with welding for ferrous materials. Types and dimensions of solder-joint fittings are given in **Table 16.** The clearance between the two surfaces is maintained within narrow limits. The solder, with suitable flux, is run into this narrow annulus in a molten condition. The joint depends on the large area of contact, and of shear. **Hard solder** is ordinarily a silver base alloy, melting above 1,000 F, whereas **soft solder** is ordinarily a tin base alloy, melting below 500 F. Hard soldering is also called **silver brazing,** under which name current practice was reviewed by Edelson.[1] Most shop soldered joints in refrigerant piping are hard soldered. The Piping Code requires the use of hard solder for field joints in refrigerant lines, except for tubes $\frac{1}{2}$-in. nominal size and less in systems containing less than 20 lb of refrigerant.

e. Flared compression fittings are permitted by the Piping Code for refrigerant lines not over $\frac{3}{4}$ in. outside diameter, using annealed copper tubing, providing all such joints are exposed for visual inspection.

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[1] Edelson, *Refrig. Eng.*, Jan. 1941.

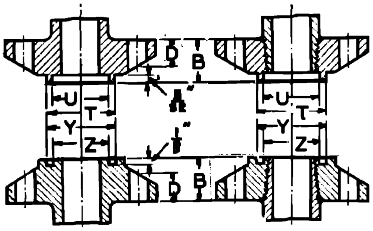


Table 12a. Standard Facing Dimensions

Nominal pipe size	Tongue			Groove			Flange thickness, min		Bolt
	Height	Outside diameter T	Inside diameter U	Depth	Outside diameter Y	Inside diameter Z	D	B	Spacing
Oval									
$\frac{1}{4}$	$\frac{5}{32}$	$1\frac{3}{8}$	1	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{1}{2}$	$\frac{13}{16}$	$2\frac{1}{4}$
$\frac{3}{8}$	$\frac{5}{32}$	$1\frac{3}{8}$	1	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{9}{16}$	$\frac{15}{16}$	$2\frac{1}{2}$
$\frac{1}{2}$	$\frac{5}{32}$	$1\frac{3}{8}$	1	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{5}{8}$	1	$2\frac{3}{4}$
$\frac{3}{4}$	$\frac{5}{32}$	$1\frac{11}{16}$	$1\frac{5}{16}$	$\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{11}{16}$	$1\frac{1}{8}$	3
Square									
1	$\frac{5}{32}$	$1\frac{7}{8}$	$1\frac{1}{2}$	$\frac{1}{8}$	$1\frac{15}{16}$	$1\frac{7}{16}$	—	$\frac{15}{16}$	$2\frac{5}{16}$
$1\frac{1}{4}$	$\frac{5}{32}$	$2\frac{1}{4}$	$1\frac{7}{8}$	$\frac{1}{8}$	$2\frac{5}{16}$	$1\frac{13}{16}$	—	$1\frac{1}{16}$	$2\frac{5}{8}$
$1\frac{1}{2}$	$\frac{5}{32}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$\frac{1}{8}$	$2\frac{9}{16}$	$2\frac{1}{16}$	—	$1\frac{1}{8}$	$2\frac{13}{16}$
2	$\frac{5}{32}$	$3\frac{1}{4}$	$2\frac{7}{8}$	$\frac{1}{8}$	$3\frac{5}{16}$	$2\frac{13}{16}$	—	$1\frac{1}{4}$	$3\frac{7}{16}$
$2\frac{1}{2}$	$\frac{5}{32}$	$3\frac{3}{4}$	$3\frac{3}{8}$	$\frac{1}{8}$	$3\frac{13}{16}$	$3\frac{5}{16}$	—	$1\frac{3}{8}$	$3\frac{7}{8}$
3	$\frac{5}{32}$	$4\frac{3}{8}$	$4\frac{1}{4}$	$\frac{1}{8}$	$4\frac{11}{16}$	$4\frac{3}{16}$	—	$1\frac{1}{2}$	$4\frac{5}{16}$
$3\frac{1}{2}$	$\frac{5}{32}$	$5\frac{1}{4}$	$4\frac{3}{4}$	$\frac{1}{8}$	$5\frac{3}{16}$	$4\frac{11}{16}$	—	$1\frac{5}{8}$	$4\frac{3}{4}$
4	$\frac{5}{32}$	$5\frac{11}{16}$	$5\frac{3}{16}$	$\frac{1}{8}$	$5\frac{3}{4}$	$5\frac{1}{8}$	—	$1\frac{3}{4}$	$5\frac{1}{8}$

All dimensions are given in inches.

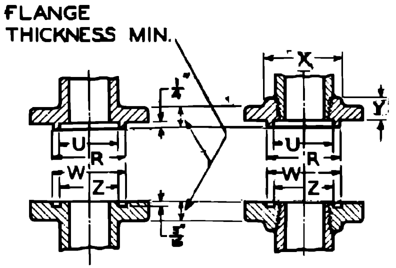
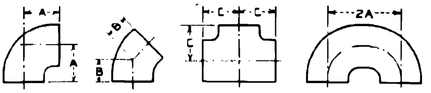


Table 12b. Standard Facing Dimensions (Round Flanges)

Nominal pipe size	Tongue			Groove			Flange thickness, min		Diameter of bolt circle	Diameter hub, min X	Length through hub, min Y
	Height	Outside diameter R	Inside diameter U	Depth	Outside diameter W	Inside diameter Z	Tongue	Groove			
5	$\frac{1}{4}$	$7\frac{5}{16}$	$6\frac{5}{16}$	$\frac{3}{16}$	$7\frac{3}{8}$	$6\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{9}{16}$	$9\frac{1}{4}$	7	$1\frac{7}{8}$
6	$\frac{1}{4}$	$8\frac{1}{2}$	$7\frac{1}{2}$	$\frac{3}{16}$	$8\frac{9}{16}$	$8\frac{7}{16}$	$1\frac{11}{16}$	$1\frac{5}{8}$	$10\frac{5}{8}$	$8\frac{1}{8}$	$1\frac{5}{16}$
8	$\frac{1}{4}$	$10\frac{5}{8}$	$9\frac{3}{8}$	$\frac{3}{16}$	$10\frac{11}{16}$	$9\frac{5}{16}$	$1\frac{7}{8}$	$1\frac{13}{16}$	13	$10\frac{1}{4}$	$2\frac{3}{16}$
10	$\frac{1}{4}$	$12\frac{3}{4}$	$11\frac{1}{4}$	$\frac{3}{16}$	$12\frac{13}{16}$	$11\frac{3}{16}$	$2\frac{1}{8}$	$2\frac{1}{16}$	$15\frac{1}{4}$	$12\frac{5}{8}$	$2\frac{3}{8}$
12	$\frac{1}{4}$	15	$13\frac{1}{2}$	$\frac{3}{16}$	$15\frac{1}{16}$	$13\frac{7}{16}$	$2\frac{1}{4}$	$2\frac{3}{16}$	$17\frac{3}{4}$	$14\frac{3}{4}$	$2\frac{9}{16}$

All dimensions are given in inches.

Table 13. Dimensions of Butt Welding Fittings (ASA B16.9-1940), inches



Nominal size	A	B	C
$\frac{3}{4}$*	$1\frac{1}{8}$	—	—
1	$1\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{2}$
$1\frac{1}{4}$	$1\frac{7}{8}$	1	$1\frac{7}{8}$
$1\frac{1}{2}$	$2\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{4}$
2	3	$1\frac{3}{8}$	$2\frac{1}{2}$
$2\frac{1}{2}$	$3\frac{3}{4}$	$1\frac{3}{4}$	3
3	$4\frac{1}{2}$	2	$3\frac{3}{8}$
$3\frac{1}{2}$	$5\frac{1}{4}$	$2\frac{1}{4}$	$3\frac{3}{4}$
4	6	$2\frac{1}{2}$	$4\frac{1}{8}$
5	$7\frac{1}{2}$	$3\frac{1}{8}$	$4\frac{7}{8}$
6	9	$3\frac{3}{4}$	$5\frac{5}{8}$
8	12	5	7
10	15	$6\frac{1}{4}$	$8\frac{1}{2}$
12	18	$7\frac{1}{2}$	10

Thickness of metal conforms to either Schedule-40 or Schedule-80 pipe.

* Not covered in ASA 16.9-1940.

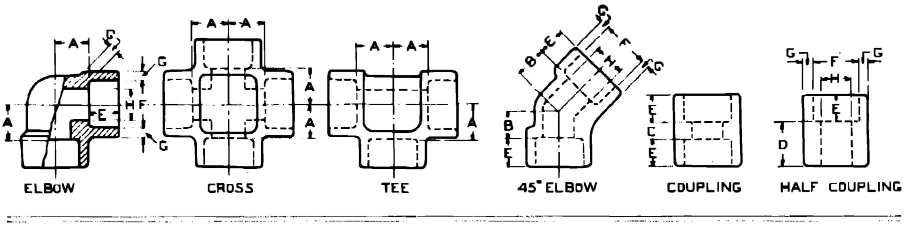
Dimensions for these fittings are given in **Table 17.**

5. Expansion and contraction of pipes take place with changes of temperature, and the piping system must be physically flexible enough to take up these changes in length by deformation, without being unduly stressed. The amount of such change in length, in inches per 100 linear feet, is given in **Table 18.** The Piping Code provides that slip-type expansion joints shall not be used with refrigerants, but that flexibility shall be provided by pipe bends, offsets, or changes in direction of the pipe line itself, also that supports shall be so arranged that they do not unduly restrain the thermal expansion or contraction of the pipe.

So long as the piping system is in one plane from anchor point to anchor point, changes of length must be taken up in bending the pipe, but if, as is usually the case, it is possible to run the pipe in three dimensions, a large part of the deformation can be taken up in torsion. The force required to deform the pipe must be supplied by the anchor points, which must be amply strong for this. These anchor points are ordinarily the equipment to which the pipe is attached. The more flexible the system,

Table 14. Dimensions of Steel Socket Welding Fittings

(Proposed ASA B16.11)*



Nominal pipe size	A	B	C	D	E	F	G Sch. 40	G Sch. 80	H Sch. 40	H Sch. 80
$\frac{1}{8}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	0.420	0.125	0.125	0.269	0.215
$\frac{1}{4}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{3}{8}$	0.555	0.125	0.139	0.364	0.302
$\frac{3}{8}$	$\frac{17}{32}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{11}{16}$	$\frac{3}{8}$	0.690	0.125	0.158	0.493	0.432
$\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	0.855	0.136	0.184	0.622	0.546
$\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{1}{2}$	1.065	0.141	0.193	0.824	0.742
1	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{1}{2}$	1.330	0.166	0.224	1.049	0.957
$1\frac{1}{4}$	$1\frac{1}{16}$	$\frac{11}{16}$	$\frac{1}{2}$	$1\frac{3}{16}$	$\frac{1}{2}$	1.675	0.175	0.239	1.380	1.278
$1\frac{1}{2}$	$1\frac{1}{4}$	$\frac{13}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{2}$	1.915	0.181	0.250	1.610	1.500
2	$1\frac{1}{2}$	1	$\frac{3}{4}$	$1\frac{5}{8}$	$\frac{5}{8}$	2.406	0.193	0.273	2.067	1.939
$2\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{8}$	$\frac{3}{4}$	$1\frac{11}{16}$	$\frac{5}{8}$	2.906	0.254	0.345	2.469	2.323
3	$2\frac{1}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$1\frac{3}{4}$	$\frac{5}{8}$	3.535	0.270	0.375	3.068	2.900

* Standard also covers Sch.-160 fittings.

the smaller the reaction on the equipment.

Since the advent of high temperatures in steam practice in the 1920's, an extensive literature on the analysis of stresses due to expansion of piping has been built up, largely in the Transactions of the ASME. However, for the relatively small changes in temperature ordinarily encountered in refrigeration, which seldom are as great as 200 F between erection and operation, the adequacy of provisions for expansion or contraction can usually be determined by inspection.

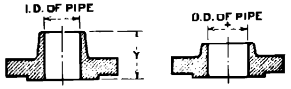
Supports for piping should be provided at intervals not exceeding 40 times the outside diameter of the pipe. In addition to supporting the weight of the pipe, including contents and insulation, it is necessary to provide **sway-bracing** to minimize vibration. If the pipe is pitched to drain, the pitch is determined by the frequency of supports, so that the deflection between supports does not introduce pockets which will not drain. Care must be taken that the supports do not unduly restrain the expansion or contraction, or introduce excessive stresses at joints or equipment, and where expansion stresses are at all critical, design of supports is equally important with providing the proper path. The Piping Code provides for certain minimum dimensions for hangers to guard against mechanical damage, corrosion, etc.

6. Protection against damage is necessary, particularly for small lines. Small insulated lines have a false appearance of strength. Where considerable traffic is adjacent, it is necessary to provide **protection** from **impact** from carelessly handled handtrucks, overhanging loads, ladders and the like. The Piping Code provides that refrigerant piping crossing an open space which affords passageway shall be not less than $7\frac{1}{2}$ ft above the floor unless against the ceiling, and that it shall not be placed in public hallways, lobbies, stairways, elevator or dumbwaiter shafts, except that it may pass across a public hallway if there are no joints in the section in the hallway, and if non-ferrous tubing 1 in. nominal size and less is contained in a rigid metal pipe. It also provides that all refrigerant pipe, with or without insulation, be exposed to view, except for mechanical protection specified in the Code, or when located in a cabinet of a unit system.

7. Control of flow may be either the regulation of the amount of flow, as by an expansion valve, or the shutting off of flow, as by a stop valve. Either may be hand-controlled or automatic. The latter may be controlled by liquid level, temperature or pressure, and is treated in Chapter 22. In any event, control is accomplished by restricting or closing the opening through which the fluid passes.

The Piping Code provides that refrigerant valves shall be of the globe, angle or needle type, gate valves and cocks being barred for refrigerants except in industrial plants having competent operating men constantly in charge. Gate valves and cocks are, however, in common use as stop valves for brine, water and other fluids. Refrigerant valves having stuffing boxes or valve

Table 15. Dimensions of Welding Flanges (ASA B16e-1939) inches

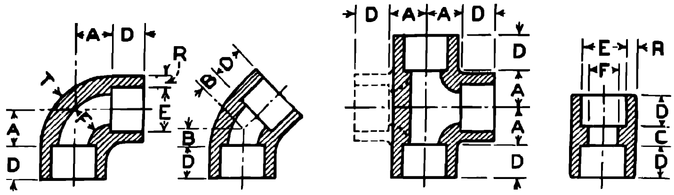


Nominal size	Y	
	150 psi	300 psi
$\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{1}{16}$
$\frac{3}{4}$	$2\frac{1}{16}$	$2\frac{1}{4}$
1	$2\frac{3}{16}$	$2\frac{7}{16}$
$1\frac{1}{4}$	$2\frac{1}{4}$	$2\frac{9}{16}$
$1\frac{1}{2}$	$2\frac{7}{16}$	$2\frac{11}{16}$
2	$2\frac{1}{2}$	$2\frac{3}{4}$
$2\frac{1}{2}$	$2\frac{3}{4}$	3
3	$2\frac{3}{4}$	$3\frac{1}{8}$
$3\frac{1}{2}$	$2\frac{13}{16}$	$3\frac{3}{16}$
4	3	$3\frac{3}{8}$
5	$3\frac{1}{2}$	$3\frac{7}{8}$
6	$3\frac{1}{2}$	$3\frac{7}{8}$
8	4	$4\frac{3}{8}$
10	4	$4\frac{5}{8}$
12	$4\frac{1}{2}$	$5\frac{1}{8}$

Outside diameter, facing, drilling and bolting of 150-psi and 300-psi steel flanges correspond respectively to 125-psi cast iron (except steel has $\frac{1}{16}$-in. raised face) and 250-psi cast iron flanges, given in Fig. 2 and Table 11.

Welding neck refrigerant flanges to match manufacturers' standards are generally available.

Table 16. Dimensions of Cast Brass Soldered Joint A40.3 - 1941), inches



Nominal size	A	B	C	D	E (min)	F	T	R
1/4	1/4	3/16	1/16	5/16	0.378	0.31	0.08	0.048
3/8	5/16	3/16	1/16	3/8	0.503	0.43	0.08	0.048
1/2	7/16	3/16	1/8	1/2	0.628	0.54	0.09	0.054
3/4	9/16	1/4	1/8	3/4	0.878	0.78	0.10	0.060
1	3/4	5/16	1/8	15/16	1.1285	1.02	0.11	0.066
1 1/4	7/8	7/16	1/8	1	1.3785	1.26	0.12	0.072
1 1/2	1	1/2	1/8	1 1/8	1.629	1.50	0.13	0.078
2	1 1/4	9/16	3/16	1 3/8	2.129	1.98	0.15	0.090
2 1/2	1 1/2		3/16	1 1/2	2.629	2.46	0.17	0.102
3	1 3/4	3/4	3/16	1 11/16	3.129	2.94	0.19	0.114
3 1/2	2	7/8	3/16	1 15/16	3.629	3.42	0.20	0.120
4	2 1/4	15/16	1/4	2 3/16	4.129	3.90	0.22	0.132
5	3 1/8	1 7/16	1/4	2 11/16	5.129	4.87	0.28	0.168
6	3 5/8	1	1/4	3 1/8	6.129	5.84	0.34	0.204

Standard also includes external shoulder fittings, reducing fittings, caps, plus bushings and fittings with pipe thread end; wrought fittings, clearances, tolerances; ratings, markings and materials.

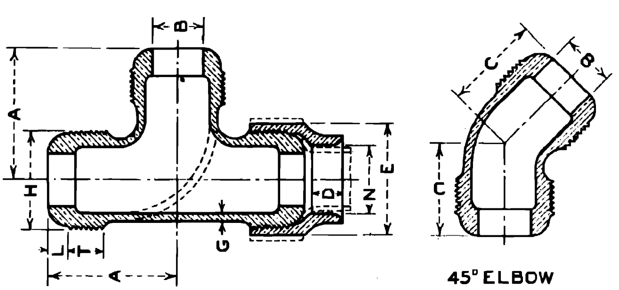


Table 17. Dimensions of Brass Fittings for Flared Copper Tubes (ASA A40.2-1936), inches

Nominal size O.D. of tube	3/8 1/2	1/2 5/8	3/4 7/8	1 1 1/8	1 1/4 1 3/8	1 1/2 1 5/8	2 2 1/8
A	1.42	1.53	1.78	2.09	2.28	2.56	3.06
B	3/8	1/2	3/4	1	1 1/4	1 1/2	2
C	1.06	1.12	1.28	1.44	1.58	1.75	2.06
D	0.19	0.25	0.38	0.50	0.63	0.75	1.00
E	1.08	1.22	1.48	1.90	2.16	2.56	3.26
G	0.09	0.09	0.10	0.11	0.12	0.13	0.15
H	0.875	1.000	1.250	1.625	1.875	2.250	2.875
L	0.25	0.26	0.28	0.30	0.32	0.34	0.38
N	0.52	0.65	0.90	1.15	1.41	1.66	2.16
T	0.43	0.43	0.50	0.56	0.62	0.69	0.81
Threads per inch	14	14	14	12	12	12	12

Standard also gives rating, material, marking, details of threading, and tolerances.

stem packing must be of the back-seating type, to permit repacking under pressure. Stop valves are required by the Code as follows:

In systems having more than 20 but less than 100 lb of refrigerant:

- a. At each inlet and each outlet pipe of each compressor.
- b. At each outlet of each liquid receiver.

In systems having 100 lb or more of refrigerant:

- a. At each inlet and each outlet pipe of each compressor.
- b. At each inlet and each outlet pipe of each liquid receiver.
- c. At each liquid and each suction branch header.

Stop valves used with soft annealed copper tubing or hard drawn copper tubing ¾ in. nominal size or smaller must be securely mounted independent of tubing fastening or supports. Stop valves placed where it is not obvious what they control must be suitably labelled.

Expansion valves are ordinarily needle valves, and the bore should be small enough to require a considerable opening of the valve for normal flow, so that adjustment is not too delicate.

Convenience for operation and maintenance is a criterion of design, and will tell the experienced engineer at a glance the competence of the designer. Valves should be so located that they are convenient to operate, joints should be easy of access for maintenance, location of lines should be such that they are easy to erect, easy to trace and present a neat appearance, and still do not obstruct or otherwise interfere with the use of adjacent space or present hazards such as low head room, projecting valve stems or the like.

Table 18. Expansion (or Contraction) of Piping in inches/100 ft from 50 F

Temperature, F	Cast iron	Wrought iron	Steel	Brass and copper
−50	−0.70	−0.76	−0.73	−1.10
−25	−0.53	−0.58	−0.56	−0.85
0	−0.36	−0.40	−0.38	−0.57
25	−0.18	−0.20	−0.19	−0.29
50	0.00	0.00	0.00	0.00
100	+0.36	+0.39	+0.38	+0.58
150	+0.74	+0.81	+0.77	+1.18
200	+1.14	+1.25	+1.19	+1.81
250	+1.56	+1.71	+1.63	+2.45
300	+1.99	+2.18	+2.09	+3.10

8. Restrictions imposed by **codes, laws and regulations** applying at the site of an installation must be known to the designer and obeyed. The Safety Code for Mechanical Refrigeration and the Code for Pressure Piping are primarily drawn up as guides for safe practice, and should be followed whether they have been given the force of law in a particular community or not. Their principal requirements as they apply to refrigeration piping are practically identical and are outlined in this chapter. They are the basis of most Municipal and State Codes, and will become more so as time goes on, but some ordinances go beyond them in requiring heavier piping and in other special features, and it is necessary for the designer to become familiar with the specific requirements applying at the particular place where the piping is to be installed.

17. OILS AND LUBRICATION

ORDINARILY, the selection of the proper lubricating oil for a given type of refrigeration equipment will be the responsibility of the manufacturer. Only the manufacturer has the necessary laboratory resources and test facilities to make a proper evaluation of all the necessary factors. Servicing of refrigeration equipment should follow manufacturers' recommendations; it is only in extraordinary instances that the individual engineer will find it necessary to make the selection. Therefore, this chapter is written primarily from the standpoint of the manufacturer, giving the basic reasons affecting the original choice.

In general, refrigeration equipment falls into two classes: (1) compressors with shaft seal and external motor drive; and (2) the hermetically sealed unit compressor with direct shaft drive and internal motor operating directly in the oil and refrigerant mixture. The latter type is distinguished by having the motor heat liberated within the compressor shell, whereas in the former type the motor heat is liberated externally. In the shaft seal type, oil temperatures seldom exceed 120 F; in the hermetically sealed type, they may reach 225 F under severe operating conditions. Furthermore, in the shaft seal type of compressor it is relatively easy to pump down the unit, remove the oil charge and replace with fresh dehydrated oil. For systems using Freon refrigerants, the oil charge is usually replaced whenever there is danger of contamination due to opening the system or other cause, whereas on systems using ammonia as a refrigerant, it is customary to replace the oil periodically.

Since the hermetically sealed unit compressor is welded in steel, it is not an easy matter to remove the refrigerant charge and remove the oil; therefore it is the usual practice to consider the original oil charge as the lubricant for the operating life of the machine. These types of machines are usually considered to have a minimum operating life of at least ten years and this consideration, together with the high operating temperatures, has placed entirely new demands upon the chemical stability of the lubricant used.

Furthermore, the design of some types of hermetically sealed units, such as the single vane rotary, requires the lubricant to perform much more efficiently as a sealing agent. In the reciprocating design, the lubricant film is spread over the entire area of contact between the piston and the cylinder wall, so that there is a very large sealing area to resist leakage of refrigerant from the high to the low-pressure side. In the single vane rotary type, however, the only sealing area is a line contact between a vane and a roller, yet this oil film is expected to prevent leakage just as efficiently as in the older type of reciprocating compressors. When it is borne in mind that in all instances, the viscosity and sealing power of oil are diluted to some extent by solution of the refrigerant in the oil, it can readily be seen that these requirements are very stringent.

In the past, refrigeration grade lubricating oils have been almost universally derived from naphthenic base crudes. They have been well refined pale oils characterized by good stability, low carbon residue, low pour point, medium flash and fire, and low viscosity index. They have been satisfactory with many millions of mechanical refrigeration units. In recent years, however, the extreme stability demands associated with the design of the hermetically sealed compressor unit, as outlined above, have required improvements in certain characteristics which were not inherently available in these naphthenic oils. Specifically, the demand has arisen for a higher viscosity index, greater chemical stability, and lower solvent power for the halogenated type of refrigerant. These properties, and particularly the property of high viscosity index, could only be obtained from crudes having a high paraffin content. Oils derived from such crudes, when compared to the refrigeration oils derived from the naphthenic base crudes, have always been characterized by relatively high wax content and a high pour point. However, it has been found possible to overcome in great measure the handicap of a high pour point and to develop oils for refrigeration purposes which have had very successful use, particularly where extremely low temperatures are not encountered.

Some authorities object to designating these oils as high viscosity index oils, insisting that the property of high viscosity index is purely coincidental, and that their most desired property is chemical stability and resistance to high operating temperatures. However, they have come to be known as high viscosity index oils, and will be so designated in this discussion.

1. Selection of Type of Oil

Table 1 shows a complete summary of the properties of the two types of oils previously discussed. Those with long experience will recognize, however, that it is impossible to select the proper lubricating oil for a refrigeration system on the basis of laboratory data. Successful refrigeration engineers insist on a complete series of **performance tests,** since changes in mechanical design often have an unsuspected influence on the behavior of the lubricant. It is also advisable to conduct a long term life test on the oil, in the unit in which it is to be used, under severe operating conditions. A representative test of this kind is a life test at 110 F ambient temperature for a period of two years. During this time, the unit is carefully checked at frequent intervals for efficiency of operation, and at the conclusion of the test, it is torn down and carefully inspected for gum, varnish, sludge, corrosion and copper plating, and wax deposits. With this information available, the final selection can then be made with a considerable degree of confidence.

In this final selection, **cost** will be an important item. The high viscosity index, high stability oils may be somewhat more expensive than those of low viscosity index

Table 1. Refrigeration Oil Properties of Naphthenic Base Oils

Manufacturers' Brand No.*	1	2	3	4	5	6
Specific gravity at 60 F	.902–.913	.904–.916	.913–.925	.919–.932	.923–.937	.929–.914
Flash point, ASTM open cup, F	275–285	300–310	330–340	350–360	370–380	380–395
Fire point, F	315–325	345–355	380–390	410–420	430–445	440–460
SU viscosity at 100 F, sec	75–80	100–115	150–160	280–300	500–600	750–800
Color, NPA	1–1½	1–1½	1½–2	1½–2	1½–2	1¾–2¼
Neut. no ASTM Mg KOH per gm	0.00	0.00	0.00	0.00	0.00	0.00
Pour test, ASTM F, max	−45	−40	−35	−25	−10	−5
Sligh oxidation	10–20	10–20	10–20	10–20	5–15	5–15
Steam emulsion, sec	20–40	20–40	30–60	40–80	60–100	70–110
Dielectric strength, kv min	25	25	25	25	25	25

High Viscosity Index Refrigeration Oils

Manufacturers' Brand No.*	3	4	5
Gravity, API (ASTM D287-39)	30–32.5	29–31	28–30
Flash, COC, F (ASTM D92-45)	385–400	430–450	455–470
Fire, COC, F (ASTM D92-45)	445–460	490–515	515–545
Pour point, F (ASTM D97-39)	0–5	0–5	0–5
Viscosity, SSU @ 100 F (ASTM D88-44)	145–160	300–315	490–545
Viscosity index	90–95	90–95	90–95

* Refers to viscosity grade. For example, No. 3 oil is 150 seconds SUV @ 100 F.

derived from naphthenic crudes, because of the inhibitors used and additional processing costs. It is unlikely that they will displace those oils from applications where they have performed with entire satisfaction in the past. Under severe operating conditions, the higher cost high viscosity index oils may show an actual saving when all factors are considered.

2. Selection of Oil Viscosity

The selection of the proper oil viscosity for use in a refrigerating compressor depends upon a number of factors, such as the mechanical **design of the compressor,** the **type of refrigerant** used, whether water-cooled or air-cooled, as well as the **type of service.** The lubricating oil in a refrigerating compressor performs the functions of sealing and lubrication. In most instances, bearing loads in refrigerating compressors are not high and the lubrication function is relatively easy to perform. The problem of sealing is not always so easy to accomplish. It depends to a considerable extent on the pressure differences involved and the solubility of the refrigerant in the lubricating oil.

Some refrigerants, particularly the halogenated hydrocarbon derivatives, are completely miscible with oil in all proportions. Others, such as sulfur dioxide, Freon-114 and Freon-22, are completely miscible above critical solution temperatures but are only partially miscible at lower temperatures. Since the tendency of the refrigerant to dissolve in the oil determines the viscosity of the resultant solution, the choice of the initial viscosity of the lubricating oil must be governed accordingly.

As a general rule, it can be stated that an oil should be chosen of the **lowest viscosity** which will give the necessary **sealing properties** with the refrigerant used and over the range of temperatures and pressures encountered. For these reasons, it is desirable to know the actual viscosity of the oil-refrigerant mixture at operating temperatures. Studies have been made by Rutledge[1] on several oils and refrigerants, and the data for oil and Freon-12 are shown in the curves in **Figs. 10, 11 and 12.** Rutledge does not state the type of oil

Table 2. Absorption of Ammonia in Oil

(Volume of gas absorbed by unit volume of oil)

Pressure, psi	Temperature		
	32 F	70 F	150 F
14.7	3.14	2.20	1.24
29.3	6.34	4.44	2.39
44.0	10.05	6.60	3.56
58.7	15.66	8.53	4.69
146.7	—	—	12.43

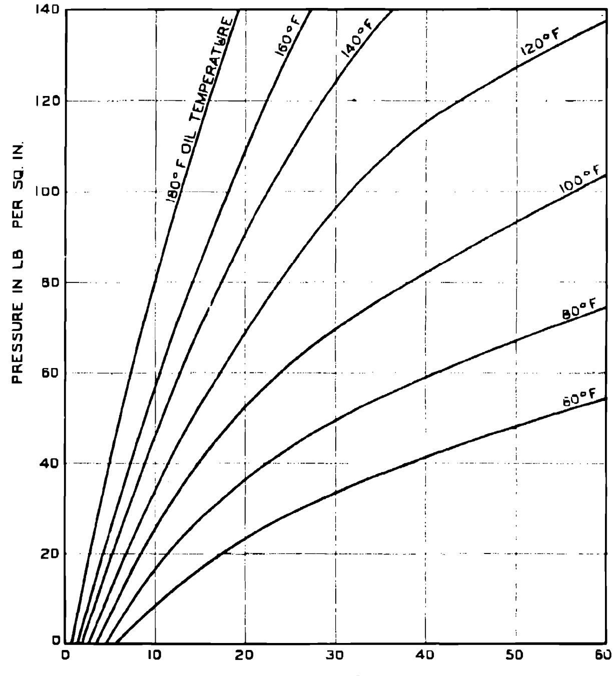


Fig. 1. Solubility of Refrigerant F-12 in Oil as a Function of Oil Temperature and Refrigerant Pressure

Table 3. Lubrication Recommendations

A. Small Systems

Refrigerant	Type of compressor	SU at 100 F oil viscosity	Typical brand (Table 1)
Ammonia	Reciprocating	150–300	3, 4
Carbon dioxide	Reciprocating	280–300	4
Sulfur dioxide	Reciprocating	70–200	1, 2, 3
Sulfur dioxide	Rotary	280–300	4
Methyl chloride	Reciprocating	280–300	4
Methylene chloride (Carrene No. 1)	Centrifugal	280–300	4
Methylene chloride (Carrene No. 1)	Rotary	150–300	3, 4
Freon-11 (Carrene No. 2)	Centrifugal	280–300	4
Freon-12	Centrifugal and reciprocating	280–300	4
Freon-21	Reciprocating	280–300	4
Freon-113	Centrifugal	280–300	4
Freon-114	Rotary	280–300	4

B. Industrial Refrigeration

(Ammonia and carbon dioxide compressors with splash, force-feed or gravity circulating systems)

Type of compressor	SU viscosity range	Typical brand (Table 1)
Where oil may enter refrigeration system or compressor cylinders	150–300 at 100 F	3, 4
Where oil is prevented from entering system or cylinders:		
In force feed or gravity systems	500–600 at 100 F	5
In splash systems	150–160 at 100 F	3
Steam-driven compressor cylinders when condensate is reclaimed for ice making	140–165 at 210 F	Pittsburgh cylinder oil

C. Miscellaneous Equipment

Type of requirement	SU viscosity at 100 F range	Typical brand (Table 1)
Bearings:		
Ring oiled, normal temperature	280–300	4
Ring oiled, low temperature	100–115	2
Chain oiled	280–300	4
Ball and roller bearings:		
Oil lubricated	280–300	4
Grease lubricated	—	No 843 roller-bearing grease
Wick oilers	280–600	4, 5

used in these tests, but it is evident from plotting the viscosity data given, that they are of the napthenic type. His curves represent average values, and he states: "There is an appreciable variation in the solubility of the different oils, and the variation from the average may be as much as ±10%." Additional data on oil-Freon-12 solubility and viscosity are shown in **Figs. 1, 8,** and **9**.

A practical method for determining the minimum safe viscosity is to determine the total volumetric efficiency of a given compressor system, using several oils of widely varying viscosity. Choose the oil of lowest viscosity giving a satisfactory volumetric efficiency. It is necessary to cite certain precautions in making these tests. They should be run at various ambient temperatures; for example, 70, 90, and 110 F. It has been observed that high viscosity index oils sometimes give as much as 5% more volumetric capacity at 110 F ambient conditions than do low viscosity index oils

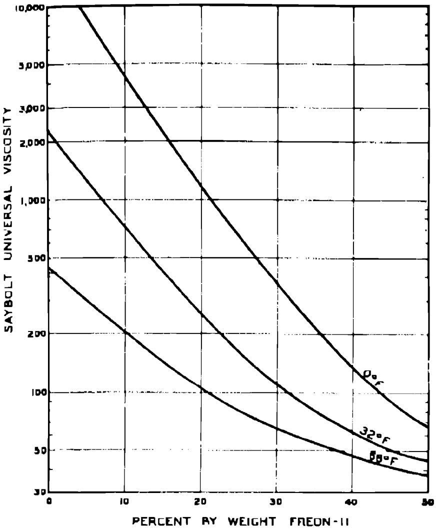


Fig. 2. Viscosity of Oils of Fig. 1 with Various Percentages of Freon-11

having a similar viscosity rating at 100 F.

The viscosity index of an oil is an indication of its tendency to reduce viscosity with increasing temperature. A low viscosity index oil may have an index as low as 0, a high viscosity index may be 100. **Fig. 16** shows the change of viscosity with temperature of two oils having viscosity indices of 0 and 92, but identical viscosity of 500 seconds S.U.V. at 100 F. Once the viscosity of the oil has been chosen, many other considerations become important, some of which are not covered by the usual lubricating oil specifications. These other considerations may be considered point by point and the governing factors outlined.

Figs. 10, 11, and 12 give the **viscosity** and **dilution ratio characteristics** of oils with Freon-12. If the temperature and pressure of the system are known, Fig. 11 will give the composition by volume of the liquid phase. Using this figure and the value of the viscosity of the pure oil at the given temperature, Fig. 10 gives the viscosity of the mixture and an approximate value for the **specific gravity.** The **absolute viscosity** in centipoises may then be obtained by multiplying the kinematic viscosity in centistokes by the specific gravity.

A numerical example follows: Given an oil whose viscosity at 80 F is 450 Saybolt Seconds, and operating conditions of 40 psi Freon pressure at this same temperature. On Fig. 11 the volume ratio is given as 0.12. Fig. 10 shows that when the oil is diluted to this extent its viscosity is reduced to 105 Saybolt Seconds or 22 centistokes. Since the specific gravity is increased from 0.88 to 0.92, this is equivalent to 20.2 centiposes absolute viscosity.

3. Pour Test and Cloud Point

It has been generally considered in the past that oils for refrigeration application require low pour test and low cloud point. Recent developments have indicated that this is not categorically true. In the past, most refrigeration oils have been refined naphthenic base stocks. These naphthenic base oils have lower cloud and pour points than paraffinic or Mid-continent base oils and usually do not require dewaxing for lubrication applications. Paraffinic or Mid-continent base oils have high wax contents

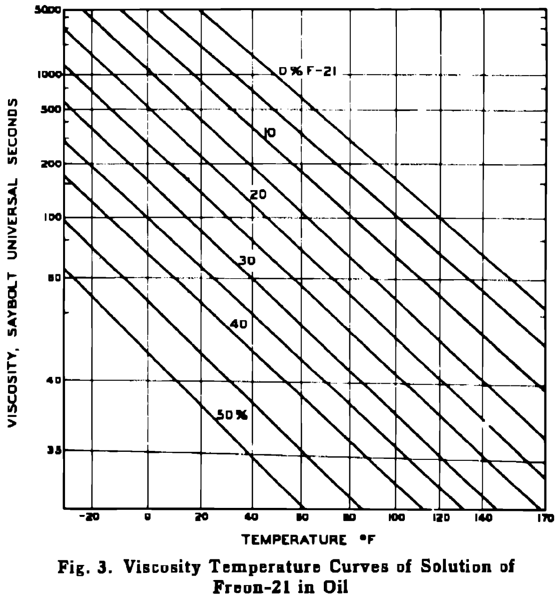


Fig. 3. Viscosity Temperature Curves of Solution of Freon-21 in Oil

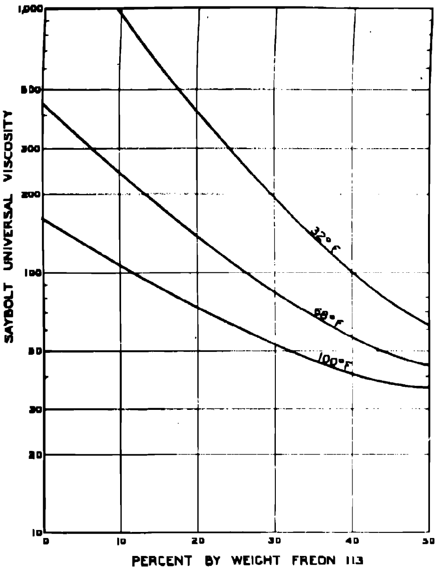


Fig. 4. Viscosity with Various Percentages of Freon-113

and require considerable dewaxing in order to produce lubricants that are satisfactory for refrigeration use. The advent of the modern dewaxing processes, however, has made it practical to dewax such oils very satisfactorily. However, the suitability of such oils for refrigeration applications cannot be judged entirely by the usual criteria of cloud point, pour point, and so-called floc point.

It should be pointed out that in a refrigerating compressor some oil is always circulated with the refrigerant, but in a well designed compressor this quantity usually does not exceed 1 or 2% of the refrigerant circulated, and in almost no instance does it exceed 10%. Consequently, it is the pour point or the cloud point of the lubricating oil mixture corresponding to that actually circulated that becomes the important criterion. It has even been found in some instances that a cloud point or a floc point of the mixture does not have too much bearing on the actual performance in the compressor. It has been observed that some dewaxed oils show a relatively high cloud point in the presence of refrigerants, but that the material in the oil that formed the cloudiness is of such a nature that it redissolves in the refrigerant very readily on a slight increase of temperature. On the other hand, some oils which show a low cloud point or floc point, separate a hard wax which is very difficult to redissolve.

Mechanical design also affects the susceptibility of a given equipment design to wax deposition. Experience has shown that wax is desposited from refrigerant solu-

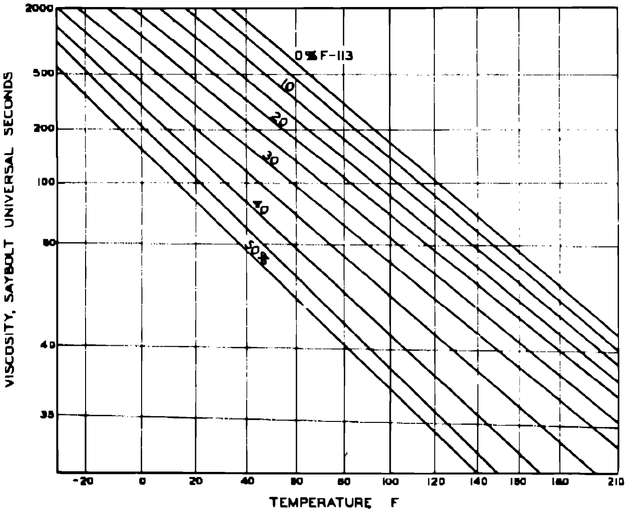


Fig. 5. Viscosity Temperature Curves of Solution of Freon-113 in Oil

tions at points where sharp bends occur, and the suspended wax particles build up on the walls of the tubing by impingement. Careful designing to avoid such bends can materially lessen the tendency to wax deposition.

Modern refrigeration equipment covers a wide range of operating temperatures. At the high end of the scale, air conditioning equipment operates at low side temperatures, usually above the freezing point. At the low end of the scale, equipment for special testing, such as low temperature Arctic weather rooms, operates at temperatures as low as −100 F. Obviously, pour point and cloud point may have no bearing on the selection of an oil for one of the air conditioning applications, while the amount and character of wax indicated by a cloud point or floc point test may greatly influence the choice of lubricant where extremely low operating temperatures are encountered.

4. Volatility

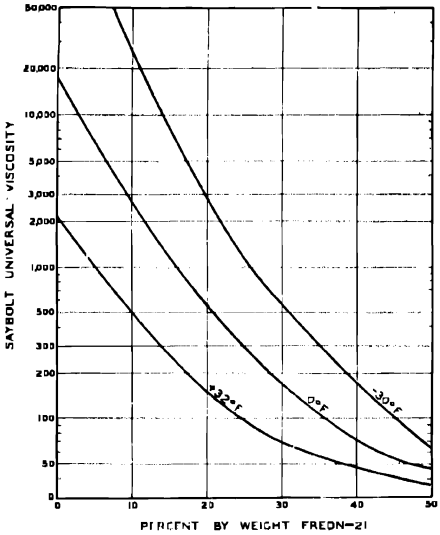


Fig. 6. Viscosity of Oil with Various Percentages of Freon-21

Volatility of lubrication oils should be such that they will not vaporize at pressures and discharge temperatures involved in the high side of the system. Since many compressor systems are dehydrated under vacuum while containing small quantities of oil, it is also important that the oils should not be too volatile under these conditions.

Flash and **fire point** of oils used in refrigerating lubricants are not too important with most refrigerants. However, some refrigerants, such as sulfur dioxide, ammonia, and methyl chloride, have a high

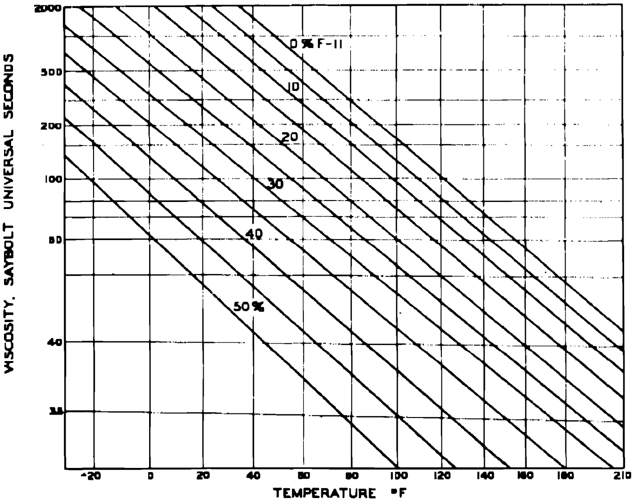


Fig. 7. Viscosity Temperature Curves of Solution of Freon-11 in Oil

ratio of specific heats $(C_p)/(C_v)$ and consequently have a high temperature of adiabatic compression. Such refrigerants frequently produce carbonization of oils with low flash and fire point when operating in high ambient temperatures.

5. Moisture

It is important that the moisture content of refrigeration oils be low. Moisture in refrigeration systems cannot be tolerated, since it not only produces freeze-ups at the point of expansion but also accelerates destructive chemical changes in the refrigeration system. It is therefore necessary to require extremely low moisture contents in all materials entering into the refrigeration system. The moisture content specified for refrigeration oil usually requires 20 parts per million of water or less. When oils have once been dehydrated so that they will meet this specification requirement, it is necessary to protect them from humid atmospheres and to handle them in sealed containers.

In refrigeration manufacturing plants, oils can be dehydrated to low moisture levels by several methods. The older one, known as **"blotter pressing,"** involves filtering the oil through carefully dehydrated filter paper. Some manufacturers also use **centrifuging** as a means of removing water. Another method is to expose the oil to high vacuum in thin films, which not only dehydrates the oil, but deaerates it as well, thereby introducing less air into the refrigeration system. In many instances, however, where refrigeration repair work is done in the field, it is not always possible to protect the oil from moisture contamination; in such cases it is advisable to dehydrate the system by the use of a satisfactory refrigeration **dehydrating cartridge.** Such cartridges contain efficient desiccating materials and will lower the moisture content of both refrigerant and oil to a level of about ten parts of water per million parts of oil and refrigerant.

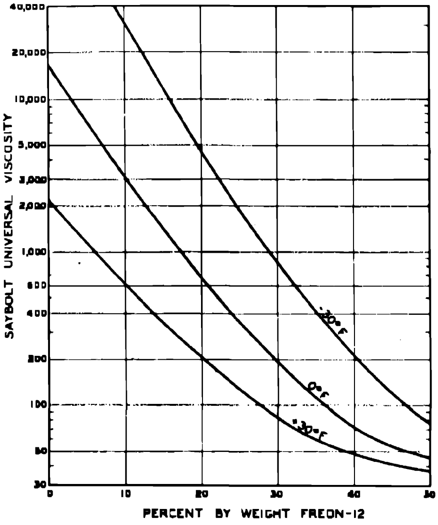


Fig. 8. Viscosity of Oil with Various Percentages of Freon-12

6. Stability

Refrigeration oils must be stable to **oxidation** and **heat.** The most severe oxidation exposures are found in the processing of the original equipment in the manufacturing plant. Under actual service conditions, oxidation is not a great factor in the deterioration of refrigeration oils. In a well designed and assembled refrigeration system, oxygen will be almost completely absent due to the careful evacuation of the system before charging oil and refrigerant. Since most refrigerants are inert so far as oxidation is concerned and some, such as sulfur dioxide, are definitely reducing in character, oxidation should not take place once the system is in operation. However, with the advent of sealed compressors, an increasing degree of heat stability is required in satisfactory refrigeration oils.

The average operating temperature in refrigeration systems has shown a steady increase over the years. This trend is expected to continue, especially in air conditioning systems. Many cities have already placed stringent restrictions on using water for cooling an air conditioning system. Since the design trend is more and more toward packaged units using sealed compressor systems, it is believed that the temperature requirements may maintain this upward trend. It has been observed that these high operating temperatures produce gum and varnish formations in some types of refrigeration oils. The mechanism of such varnish formation is not

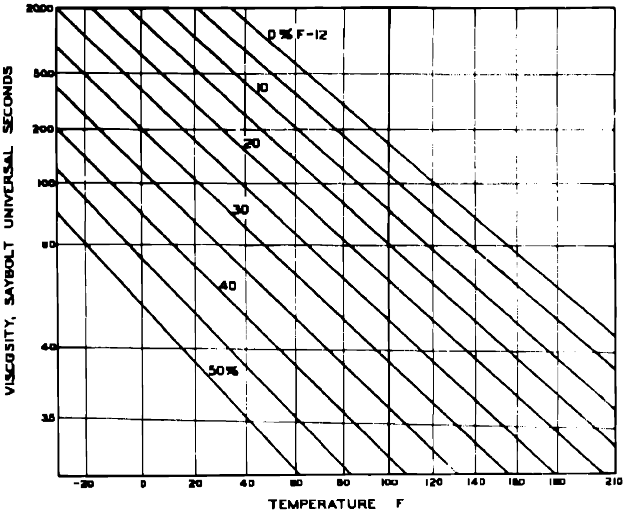


Fig. 9. Viscosity Temperature Curves of Solution of Freon-12 in Oil

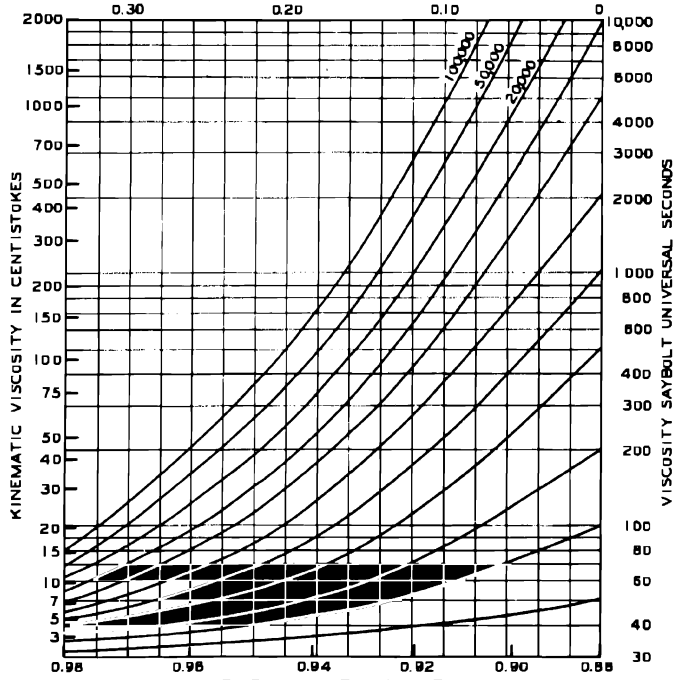


Fig. 10. Viscosity-Dilution Ratio Characteristics of Freon and Any Oil at Any Temperature—Upper Scale Shows Ratio of Freon to Oil by Volume

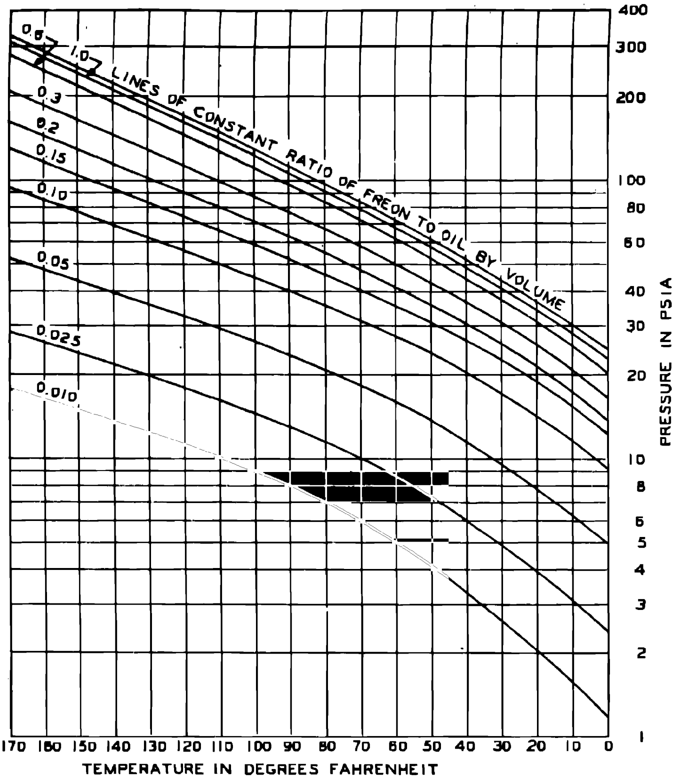


Fig. 11. Dilution Ratio Characteristics of Oil and Freon—Value of Top Line Is Infinity

clearly understood since oxidizing conditions are almost completely absent. A frequently advanced explanation is that it is caused by polymerization of unsaturates catalyzed by the presence of certain metals, particularly copper.

Much work has been done in attempting to develop chemical additives which will stabilize these oils toward gum and varnish formations, but the usual type of oxidation inhibitor has not been too effective.

7. Copper Plating

Where halogenated refrigerants have been used, a phenomenon known as "copper plating" has been observed. Every refrigeration system contains some copper. In ordinary reciprocating compressor systems, the quantity may be confined very largely to the connecting lines. In sealed rotary compressors, however, the motor windings are in direct contact with the oil and refrigerant of the system. A chemical reaction takes place whereby the copper in contact with the oil and refrigerant is dissolved and goes into solution in the oil and refrigerant where it can be detected by sensitive chemical analysis. Under suitable conditions, it then plates out on the surfaces of other metals, but most usually on steel surfaces. Furthermore, since bearing surfaces are usually the brightest, cleanest and most highly polished in the system, the deposition occurs preferentially at these points. Where tolerances are very close, the deposited copper may increase to such a thickness that binding and galling of such a surface may occur. This situation is usually not serious in the older type, slow-speed reciprocating compressors. In certain types of rotary compressors, however, where tolerances are of the order of one-tenth thousandth (0.0001) of an inch, this deposition may eventually result in the failure of the system.

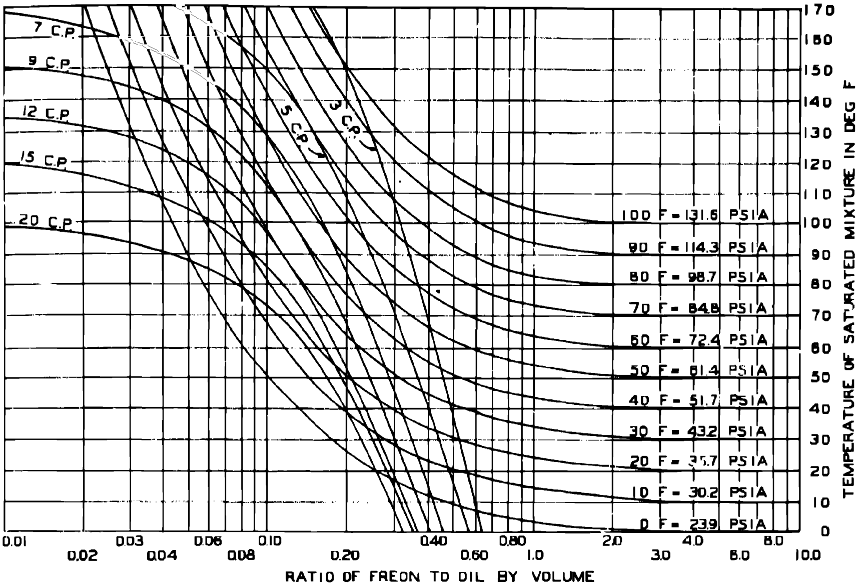


Fig. 12. Viscosity and Dilution Ratio Characteristics of Freon and a 500 Viscosity Oil in a Refrigerating System

Much study has been devoted to the causes and cure of this difficulty. It has been thought for some time that the lubricating oil plays an important role, and it has been demonstrated that different types of oil differ widely in their ability to dissolve and hold copper in solution. It has been a widely held theory that the copper so dissolved precipitates out in contact with steel surfaces by straight electromotive displacement. However, it has been observed in laboratory experiments that such deposition can actually take place on glass, and it is therefore believed that electromotive displacement is not always involved.

Recent work has also established the fact that copper plating can occur in the presence of the refrigerant alone without the presence of any lubricating oil. However, in oil and refrigerant mixtures, it is certain that the reaction takes place in the presence of some types of oil much more readily than it does in others, and there is no doubt that the lubricating oil is in many instances a contributing factor. It has also been established by much careful laboratory work that the reaction does not take place readily in completely dry systems, and that most difficulties due to copper plating can be largely eliminated by **adequate drying** during the **original manufacture** of refrigerating equipment and by the use of built-in **chemical dehydrators** as an integral part of the equipment in service.

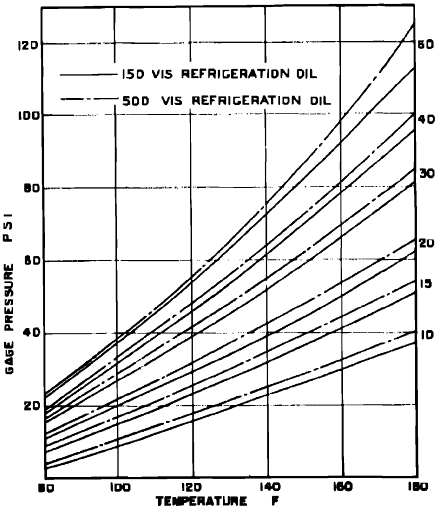


Fig. 13. Temperature Pressure Relationship of Freon-114-HVI Oil Mixtures

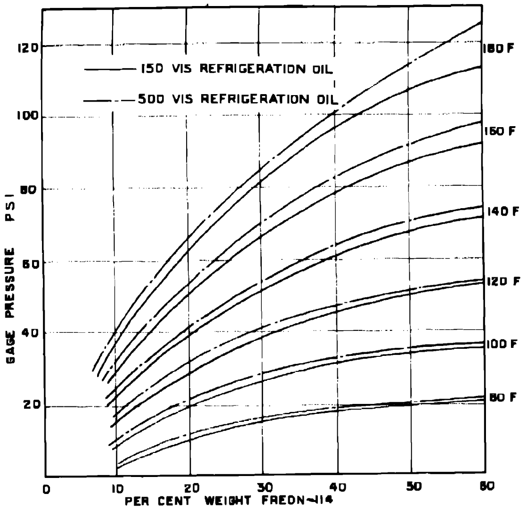


Fig. 14. Temperature-Pressure Relationship of Freon-114-HVI Oil Mixture

Due to observations with respect to copper plating in the presence of refrigerants alone, competent technical opinion is now turning to the belief that the refrigerant may be the dominant factor in the plating reaction. The theory that has been advanced is that small quantities of moisture present in the system produce a slow chemical hydrolysis of even the most stable halogenated refrigerant—even those of the well-known Freon type. In fact, microanalytical methods have demonstrated the presence of chloride and fluoride ions in corrosion deposits found in refrigeration systems. It is now believed that the ultimate solution of this problem may require even more stable halogenated refrigerants than those currently in use.

8. Mutual Solubility of Oil and Refrigerant

Refrigerants fall into two classifications under this heading: (1) those completely miscible with oil in the range of operating temperatures; and (2) those which exhibit a critical solution temperature in the operating range, i.e.—separate into two immiscible phases. Freon-12 is an example of the first class, and sulfur dioxide an example of the second. Freon-114 (tetrafluorodichloroethane) shows critical solution temperatures as high as 60 F with some oils; hence oil-F-114 mixtures are usually single-phase on the high side, and two-phase on the low side. Typical curves for these relationships are shown in Fig. 15.

The amount of refrigerant which an oil will absorb from the vapor phase at a given oil temperature and refrigerant pressure is also an important criterion of refrigerant solubility. Typical curves are shown in Figs. 1 and 16.

These solubility characteristics have an important bearing on lubricant performance. The extent to which an oil dissolves refrigerant vapor at a given temperature and back pressure determines the effective viscosity of the oil-refrigerant mixture in the machine, and hence affects its sealing power. If the oil sump is located on the low-pressure side of the machine, this characteristic also affects the amount of foaming of the oil-refrigerant mixture when the compressor starts after an off cycle.

As previously pointed out, if the critical solution temperature of the oil-refrigerant

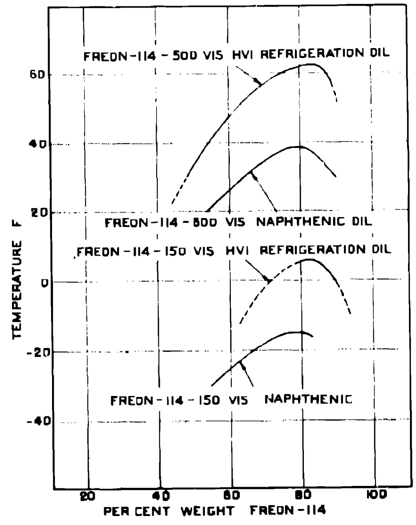


Fig. 15. Critical Solution Temperatures of Freon-114-Oil Mixtures

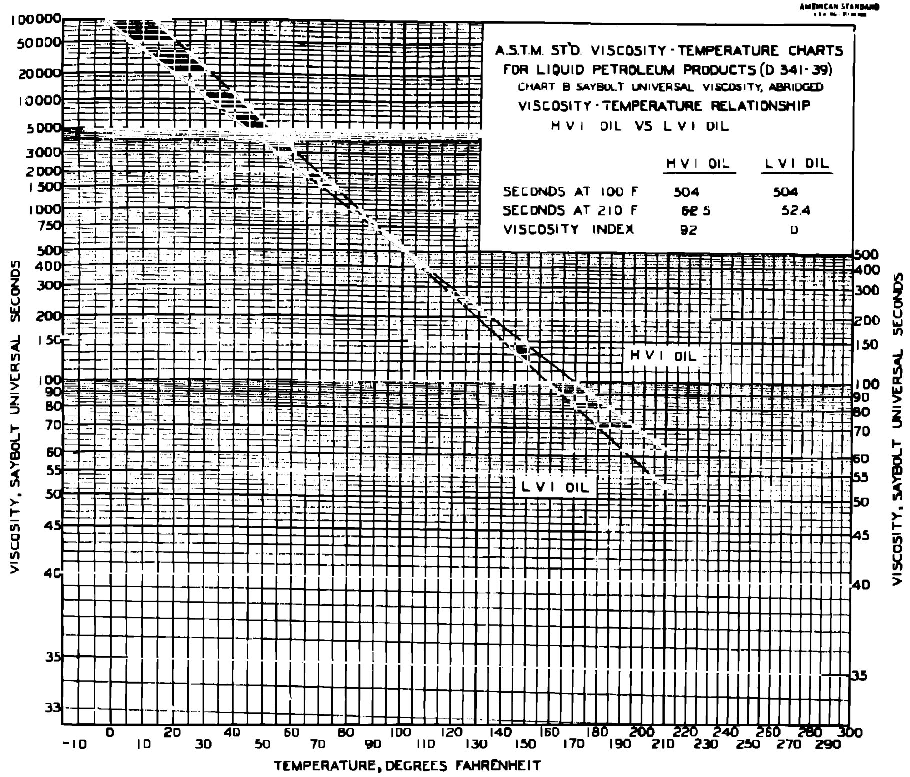


Fig. 16

mixture is appreciably higher than the evaporator operating temperature, a two-phase system may exist in the evaporator. This condition affects "oil-logging" of the evaporator, rate of oil return, and, in some instances, may affect evaporator temperatures, particularly if it is of the flooded type. In general, it may be said that high viscosity index oils show less tendency to dissolve refrigerant from the vapor phase, and exhibit higher critical solution temperatures in the liquid phase than do the low viscosity index naphthenic base oils. It must be reiterated, however, that the effect of these characteristics can only be evaluated by actual performance tests.

9. Lubrication

There are two general **types** of lubricating ststems for refrigerator compressors—those used with open type compressors and those used with closed type compressors.

a. In open type compressors (with open crankcases), the oil is withdrawn from the refrigerant system. This method of lubrication is usually found in large horizontal industrial compressors used in ice plants or industrial processes in which separate systems for the lubrication of cylinders and bearings are used. In such cases, piston and piston rod lubrication is provided by a pressure system which injects predetermined amounts of oil into the stuffing box gland and cylinder, while bearing lubrication is provided by force feed, gravity, or splash systems. A lighter oil is used for cylinders than for the lubrication of bearings where the oil is exposed to greater pressure.

Any excess of oil for cylinder lubrication should be avoided, otherwise it may form a coating in the expansion coils and decrease the efficiency of the system. To prevent this, well designed systems are provided with an oil separator on the discharge line which separates the oil entrained in the high pressure gas as it travels to the condenser.

b. In closed type compressors, the same oil serves for the lubrication of bearings and cylinders and remains in the system. A heavier grade of refrigerating oil must be used in such machines.

All industrial compressors which have been in service for some length of time should be carefully inspected for worn parts and, if advisable, heavier oils should be recommended to avoid an excess of oil being pumped into the expansion coils.

10. Industrial Compressors

Pressure lubrication is extensively used in large horizontal refrigerator compressors, while splash lubrication is better adapted to vertical compressors.

Pressure lubrication permits more accurate control of the amount of oil delivered to cylinder walls and bearings, but requires considerable piping and frequent attention by the operator, who has to refill the reservoir and regulate the flow of oil.

a. Mechanical force feed lubricators. Cylinders and bearings are extensively lubricated by mechanical lubricators consisting of a small rectangular tank, divided into several compartments and located above the parts to be lubricated. A plunger mechanism is operated from cams or levers which are connected with some rotating part of the compressor. When the compressor is in operation, the plungers pump the oil through check valves to certain points in the cylinder walls, bearings and other moving parts. The amount of oil can be regulated by adjusting the plunger stroke. Frequently different grades of oil are contained in the compartments of the tank for lubricating the compressor cylinders.

Force feed lubricators for refrigerator compressors should not be open to the air or moisture will enter the system. The function of these lubricators is to meter out the lubricant and deliver it to all points to be lubricated in minute predetermined quantities. The amount supplied by each unit can be seen in the sightglass, and any excess tends to cause carbonization at some of the hot spots frequently occurring in large industrial compressors. Care must be taken that the oil feed lines are filled before operation of the compressor begins. For this purpose a manual handcrank, which can also be operated when it is found necessary to increase the supply of oil momentarily, is usually part of the lubricator.

In large horizontal and vertical compressors with crossheads, the internal parts, cylinders, piston and rods are lubricated by mechanical lubricators with low pour test refrigerating oils. External parts such as bearings, pins, cross-head guides, governors, etc., are serviced with machine oils by an independent gravity or pressure circulating system.

In the smaller horizontal compressors, oil is introduced at the top-center of the cylinder and in double-acting compressors it is introduced in the middle of the cylinder. Automatic valves keep the oil in the feed lines by checking back pressure during the compression stroke.

In some systems the "injection method" is used. The oil is injected into the suction line and carried into the cylinders by the flow of gas, forming a film of oil over all exposed surfaces. This method is wasteful and there is always danger that excess oil will cause sticky valves.

In practically all compression machines with reciprocating parts, the oil serves also as a **seal** in the **stuffing boxes.** Hot ammonia gas has a bad effect on the packing of stuffing boxes, and sufficient lubricating oil with a high flash point should be used to prevent overheating, piston rod wear and leakage of gas. Lubrication is generally accomplished in a stuffing box by an "oil lantern" which is located between two sets of packing, and consists of hollow spaces surrounding the rod into which the oil is pumped from a reservoir. Another opening from the lantern is connected with the suction side of the compressor and any gas which leaks by the packing rings is returned to the cylinders.

In some types of double-acting compressors, the oil lantern also serves to feed the lubricating oil to the compressor cylinder. In this method, the oil penetrates past the piston rod packing and at the same time protects the packing from the bad effects of the ammonia.

Ammonia and carbon dioxide compressors are very similar to each other as far as construction and operation are concerned. The working pressure of carbon dioxide compressors is higher (from 900 to 1000 lb) and consequently the cylinder temperatures are also higher. In both types the lubricating oil must not carbonize at the high temperatures in the cylinders, must have sufficient body (viscosity) to supply a protective film on all frictional surfaces, and must readily drain into the traps at the low temperatures encountered.

Where CO_2 refrigerating machines are operated in the production of dry ice, a white oil may be required. Dry ice is frequently used in direct contact with food products and for this reason some operators insist on a water-white oil which will have no effect on the odor or color of the dry ice should an appreciable amount of oil pass the traps. To avoid such possible contamination of the dry ice, some operators are willing to sacrifice the slightly better lubricity of the pale oils.

b. Oil separators. Most industrial refrigerating systems are equipped with oil separators to aid in the separation of the oil from the refrigerant. They are generally cylindrical tanks, with baffles and a sight gage glass to indicate the level of the oil. Particles of oil mixed with the gas after it has left the compressor, are precipitated in the separator.

c. Lubricating systems. Every compressor manufacturer has his own system of lubrication which serves his equipment best. For instance, in some older types of steam driven horizontal compressors, a splash system for cylinder walls and piston lubrication is used while the external parts are oiled by oscillating mechanical lubricators. In some of the latest models oil is pumped from the crankcase into a sump, circulated under pressure and returned through filters to the crankcase. At the same time a mechanical lubricator forces engine oil to bearings and cylinders, and a third oil system is used to regulate the operation of the governor.

Another modification of the force-feed lubrication on large industrial compressors embodies a chain-driven gear pump which furnishes positive pressure lubrication. Oil from the reservoir enters the pump intake through a strainer and is discharged into a screen type filter which can be cleaned while the compressor is operating. The oil is then passed through an oil cooler and distributed through drilled passages or oil lines to the bearings, cross-head pins, guides, etc. Excess oil from the pump is bypassed to the main reservoir or to the oil sump by a relief valve. Automatic devices stop the compressor in case of oil pressure failure.

d. New compressors. A new compressor requires more oil for cylinders and bearings until the surfaces have become smooth and coated with an oil film. It is customary to break in new units with a refrigeration oil of lighter viscosity than is used in regular operation. This oil is known as "break-in" oil.

After the compressor is broken in, it is advisable to drain the oil, clean the crankcase carefully of all dirt, metallic particles, lint, etc. and refill it with clean fresh oil. After the compressor has been started again, the oil level in the reservoir should be watched and oil added until the oil level is several inches above the intake pipe, which should be marked "low level." When the compressor is shut down and the oil has drained back into the reservoir, the oil level should be marked "low level-idle" and after adding oil another level is established for "high level-idle." After the compressor is started again the resulting oil level is marked as "high level" (running.)

Frequency of oil change depends on operating conditions and time-efficiency of operation. Most manufacturers of large heavy-duty compressors recommend an oil change at least every ninety days. Instruction books furnished by each manufacturer should be followed carefully by the operating engineer.

11. Splash Lubrication

Smaller industrial and commercial vertical refrigerator compressors have enclosed crankcases. Cylinders, piston and bearings are lubricated by a splash or circulating system. In the splash system the oil (for cylinders and bearings) is stored in the crankcase, and as the crank dips into the oil during each revolution it splashes a sufficient amount of oil spray to the cylinder walls, pistons, etc. An excess of oil in the crankcase would cause violent agitation or churning of the oil by the crank, preventing precipitation of any foreign matter which might be in the system. Such excess oil might also enter the refrigerant circuit and prevent efficient operation of the condenser, evaporator and oil separator.

Refrigerating oil for splash systems must have sufficiently high viscosity at compression temperature to prevent excess oil consumption, because too light an oil will pass by the piston rings. Since air can enter commercial refrigerating systems, the oil must be resistant to oxidation and very stable, so as to reduce carbon and gum formation to a minimum.

12. Domestic and Small Systems

During the earlier stages of development of these units splash lubrication was the system most commonly used. Some trouble occurred due to "foaming" in reciprocating compressors installed in these smaller units.

Foaming is invariably caused at the start of the cycle, when return gas has been absorbed in the oil and is suddenly released as the compressor starts lowering the pressure in the crankcase. There is always danger that foam might be carried over to the high side of the compressor with the refrigerant.

This experience led to efforts to develop several **systems of pressure lubrication** in order to obtain positive circulation of the oil through the compressor. One type of pressure lubrication is similar to the lubrication of an automotive engine by an immersed gear pump in the crankcase. The pump discharges the oil under pressure (regulated by the speed of the gears) through small feed lines to the cylinder walls, or through drilled passages to the bearings. After forming the required protective film, the surplus oil returns to the crankcase by gravity and is recirculated.

Another type of pressure lubrication makes use of a rotary pump. Two blades moving in a slotted rotor, pump the oil from a reservoir through drilled passages in the motor shaft to the bearings and cylinder walls. In some units part of the oil is used to cool the motor winding before returning to the sump.

Another system applies splash lubrication to lubricate the cylinder walls. The bearings are lubricated by a reciprocating pump with a single oscillating cylinder operated from an eccentric on the crankshaft.

As stated previously, lubrication is provided for when the manufacturer charges the unit with the refrigerant and oil before it leaves the factory. Both are expected to last and give satisfactory service for years. If an oil change should be required either the manufacturer or dealer should do it, because they know the best type of refrigerating oil for their particular product.

13. Driers or Cartridge Dehydrators

Driers or cartridge dehydrators are tubular appliances, containing moisture-absorbing material such as silica gel, activated alumina, anhydrous calcium sulfate, calcium oxide or anhydrous calcium chloride, in a sack or in a dispersion tube. They are manufactured with capped or flanged-end shells which can be inserted into the liquid-circulating line. Refrigerator manufacturers and service men insert them in commercial and sealed units to remove moisture from the system. After the liquid refrigerant and the oil circulate through the cartridge for a specified time the cartridge is disconnected.

14. Lubrication of Bearings

On large heavy-duty refrigerator compressors a gear pump or motor-driven centrifugal pump forces oil to a header supplying oil to the bearings. Floor lubrication (splash system) is also used, as previously described.

Large industrial horizontal or vertical refrigerating compressors use an independent gravity pressure circulation or force feed lubricating system. In either of these systems the oil should go through a separator if it is contaminated with water.

Bearings are frequently lubricated by ring and chain oilers. A ring oiler consists of a machined ring having a larger diameter than the journal. The ring rotates with the journal, picking up oil from a reservoir and carrying it up to the bearing surface. This surface generally contains grooves for the distribution of the oil over the entire surface, after which the excess oil runs back into the reservoir. The level in the oil reservoir must be maintained sufficiently high so that the ring always dips well into the oil. Ring oilers are used for the lubrication of outboard crankshaft bearings of industrial reciprocating compressors and for rotor bearings of centrifugal compressors. Chain oilers operate on the same principle as ring oilers, except an endless chain is substituted for the ring.

Oil also acts as a shaftseal (where the shaft passes through the casing) in combination with the lapped metallic seals. This arrangement prevents air from entering the system and the loss of refrigerant to the atmosphere. In centrifugal compressors oil is supplied to the seal by pressure during operation and the metal seal faces are separated by oil pressure. When the compressor is not operating a metal-to-metal seal is made under oil.

Outboard bearings are also lubricated with sight-feed oilers consisting of a cylindrical cup filled with the oil which flows through a tube into a hole in the bearing. A glass is inserted in the oil tube so that the flow of oil is visible and can be regulated by a needle valve inserted in the tube.

15. Air Conditioning Equipment

The lubrication of small **unit type** air conditioners is similar to that previously covered under domestic and small systems. The original charge of refrigerating oil and any necessary service should be taken care of by the manufacturer or dealer from whom the equipment was purchased. In railroad, trucking, bus and airplane applications, which place more severe service on these units, regular inspection and overhaul periods are customary.

In vertical reciprocating compressors of small and medium tonnage, splash lubrication is mostly used.

In large centralized systems (large theaters, hotels, department stores, hospitals, industrial plants, office buildings, etc.), pressure lubrication is used in the large type horizontal and vertical compressors for the lubrication of cylinders and piston rods, etc. In some designs an overflow pipe from the stuffing-box oil lantern carries the oil to the suction line and delivers it to the cylinder.

Force-feed lubricators are also used. Some compressors in this service contain an enclosed type of force feed pump located on the bottom of the crankcase and driven from the main shaft. Such a pump always has positive suction and delivers oil through feed lines to all moving parts under positive pressure.

The bearings of **motors and fans** used for the distribution of the cooled air are mostly of the antifriction type, and their lubrication is identical with such bearings in industrial use. The lubricant must protect the highly polished surfaces from pitting and corrosion, must seal the housing against foreign matter (dirt and water) and must support sliding and rolling contact within the housing. Oil or grease serve as the lubricant. The recommendations of the manufacturer of such bearings, which are based on the service, operating conditions, speed and temperature, should be followed.

Before air is conditioned for circulation, dust and dirt should be removed. Various types of **filters** and **cleaners** are used to clean the air thoroughly without offering too much resistance to the air flow. Some of these filters in large air conditioning systems employ a viscous **oil film** to hold dust and dirt. The coating which gradually accumulates renders the filters inefficient unless cleaned and reoiled. Self-cleaning automatic filters are on the market in which screen panels rotate continually through an oil reservoir at the base of the filter, and the dirt is washed off and accumulated in a sludge-pan. The oil in the reservoir is changed whenever it becomes too saturated with dirt.

Reference

1. Rutledge, *Refrig. Eng.*, no. 1, vol. 35, pp. 31-36, Jan. 1938.

18. MOISTURE AND DRYING METHODS

1. Moisture in refrigerating machines constitutes a most important problem for both the manufacturer and service man. Engineering has solved most of the mechanical problems in refrigeration, while producers have virtually eliminated difficulty due to refrigerants, unless they are mishandled. In the main, satisfactory lubricating oils have been provided, reducing trouble from this source to a minimum. However, moisture is still found in machines. Absence of moisture above an allowable minimum is absolutely essential to satisfactory machine operation. It is therefore imperative that moisture be eliminated during manufacture, that entrance of moisture in a system be guarded against in all field operations and that when it does get in, removal be accomplished as speedily as possible. Moisture may be present in a system as the result of:

1. Faulty drying of equipment at factory and in service operations
2. Introduction during assembly or service operations in the field
3. Low-side leaks, resulting in entrance of moisture-laden air
4. Leakage of water condenser
5. Oxidation of certain hydrocarbons of oil, to produce moisture
6. Wet oil and/or refrigerant
7. Decomposition of motor insulation in hermetically sealed units.

Item 1 will be discussed below, and items 2 to 5 may be corrected by proper servicing. Moisture is present in such small amounts in refrigerants and oils which are bought to specification and properly handled thereafter, that virtually no difficulty arises from these sources.

Effects of Moisture

2. The presence of moisture in a refrigerating system may result in one or all of the following undesirable **effects**:

1. "Freezing up" at expansion valves or capillary tubes, ice in evaporators
2. Corrosion of metals to form sludge
3. Flapper-valve failure

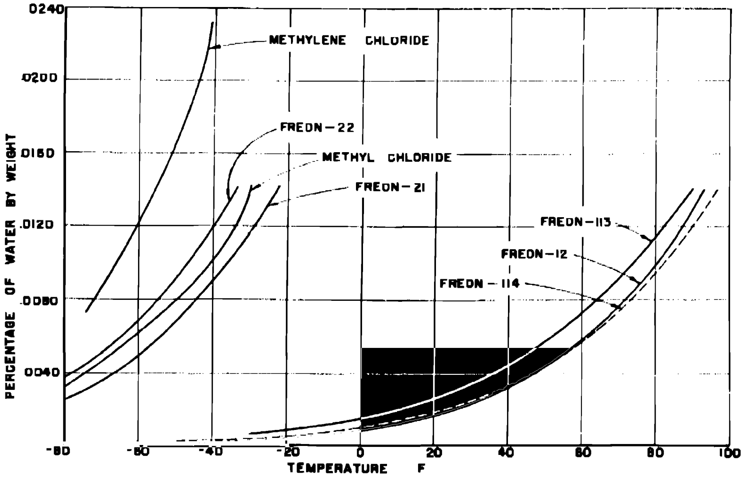


Fig. 1. Solubility of Water in Refrigerants

4. Copper plating
5. Chemical damage to insulation, etc.

Ice separates from the Freon refrigerants, methyl chloride, methylene chloride, butane, and isobutane, whenever the amount of moisture is sufficiently great and the temperature of the refrigerant low enough. The latter accounts for the formation of ice in expansion valves, capillary tubes and evaporators, but not elsewhere in the machine.

In low-temperature units a frozen expansion valve or capillary tube may result due to the separation of **wax** from oil-methyl chloride or oil-Freon mixtures.[1] The presence of wax may be demonstrated by **washing** the expansion valve with a small amount of carbon tetrachloride, and evaporating the solution so obtained. White wax or an extremely viscous oil-wax mixture proves that wax is the offending element although it does not rule out moisture. The wax separation characteristics of the oil should be determined,[1] and if the oil is not satisfactory another should be used.

The separation of **water,** as ice, is related to the solubility of water in a refrigerant. The solubility of water (ice) varies with the different refrigerants mentioned above, and is great enough in ammonia, carbon dioxide, and sulfur dioxide so that freeze-ups do not occur. The greater the solubility of water in a refrigerant, the less likely that ice will separate. **Fig.** **1**[2a] contains curves representing the solubilities of water in Freon-11, Freon-12, Freon-21, Freon-22, Freon-113, Freon-114, methyl chloride and methylene chloride. No data are available for butane and isobutane.

3. The **corrosion** of metals occurs whenever the quantity of water exceeds certain fairly well defined values.[3] **Table 1** contains the results of tests conducted on steel in glass tubes where the quantity of water could be clearly defined.

Limits for copper, brass and aluminum are slightly higher in all three refrigerants. **Aluminum** in methyl chloride is eliminated from moisture corrosion consideration, since a direct action occurs resulting in the formation of spontaneously flammable products.[4] Higher temperatures, encountered in the compressor and condenser, will lower these limits appreciably, so that corrosion occurs with less moisture in a machine at these points than with steel at room temperature.

The **mechanism** of corrosion, which results in the formation of metallic salts, (sludges) has been studied.[5] Water reacts with sulfur dioxide, methyl chloride and Freon-12 (hydrolysis[5,6]) to form sulfurous, hydrochloric, and hydrofluoric acids, respectively. These acids react with iron, copper and aluminum (see also direct action aluminum on methyl chloride) to form metallic salts (sludges) of well defined composition. By chemical analysis of these salts it is possible to demonstrate their origin as being due to water. Chemical analysis thus becomes important.

Table 1

Refrigerant	Water by weight, %	Results
Sulfur dioxide	0.03	Slight discoloration
	0.10	Slight scale
	0.15	Heavy scale
		Presence of air did not affect results
Methyl chloride	0.02	Slight discoloration
	0.03	Marked discoloration Very slight scale
	0.05	Moderate to heavy scale
		Presence of air increased corrosion in all cases
Freon-12		Similar to methyl chloride

Approximately 90% of the sludges produced in refrigerating systems are due to moisture;[5] the others are associated with oil, or minor causes.

Corrosion in a sulfur dioxide system proceeds within a few days, or even hours, provided enough water is present, whereas it is much slower for methyl chloride and Freon-12. This is due to the very rapid action of water on sulfur dioxide and the rather slow action of water on the other two refrigerants, to produce the acids responsible for corrosion.

In the presence of air, corrosion is much worse in all refrigerants except sulfur dioxide. Corrosion in a butane or isobutane system is due to the direct action of water and perhaps air, on metals.

Waring[32] has shown that a relatively large quantity of moisture in a reciprocating system using Freon-12 as refrigerant, results in accelerated flapper-valve breakage. The same system, overhauled completely and equipped with an efficient drier, operated indefinitely without flapper-valve breakage. Incipient corrosion, probably of an intercrystalline nature, was advanced as the cause of fatigue failure of the flapper-valves.

Copper plating may be the result of the action of, or related to, moisture,[7] but a proper relationship seems difficult to establish.

Consideration of the foregoing shows that the amount of water present in a refrigerating system must be small enough to avoid ice separation and corrosion. With sulfur dioxide, corrosion alone must be avoided, which probably limits the maximum water content to below 0.03%.

Some limited information[8] regarding the water content of machines that are operating satisfactorily is given in **Table 2.**

Table 2. Moisture Content of Machines Considered Sufficiently Dry for Normal Operation

Company	Refrigerant	Water by weight, %	Type of system
A.	Freon-12	.0027	Large commercial
	Freon-12	.0026	Large commercial
B.	Freon-12	.0037	Domestic hermetic
	Freon-12	.0037	Domestic hermetic
	Freon-12	.006	Domestic unit returned due to moisture
C.	CH_3Cl	.0058	Small commercial
	CH_3Cl	.0058	Small commercial
D.	CH_3Cl	.007	Small commercial
	CH_3Cl	.005	Small commercial
E.	Freon-12	.002	Hermetic
	Freon-12	.003	Hermetic
	Freon-12	.003	Small commercial
	Freon-12	.002	Hermetic
	Freon-12	.002	Small commercial
	Freon-12	.003	Hermetic

These data indicate that the moisture tolerance of Freon-12 is lower than for methyl chloride. This is quite generally borne out by experience in manufacturing and servicing. Since the quantity of moisture required to produce corrosion is well above that causing a freeze-up, elimination of freeze-ups should exclude corrosion of the type due to water.

Drying Methods

4. There are three general requisites of a satisfactory process of drying refrigerating equipment, namely, heat, time and a method for the removal of moisture.

a. **Heat,** which is necessary for the evaporation of water, may be applied by placing the equipment in an oven, using hot air or heating the apparatus with a flame, wherever possible.

b. **Time** is required for the evaporation of water, the elimination of water adsorbed on the various surfaces and the removal of the water from the equipment with dry air or by means of a vacuum.

c. In general, higher temperatures cut down, but cannot reduce the time requirement below a minimum which must be set for each process, type of equipment and the degree of drying desired. A **current of dry air** drawn or blown through the equipment removes moisture as it becomes totally or partially saturated. The rate of flow should be high enough to avoid complete saturation of the air in order to minimize the danger of deposition of moisture in some cooler portion of the equipment. On the other hand, there is little to be gained by the use of a very high rate of air flow, since the amount of water removed per unit of air will be small. Since hermetic windings can be damaged by too high temperatures or too long a time in the dehydration oven, caution should be exercised in setting up a drying procedure.

Dry air may be **manufactured** by several methods, among which are:

a. Removing the major part of the moisture with a refrigerating unit, followed by more complete drying in a unit charged with activated alumina, silica gel, or Drierite.

b. The use of a drier unit, charged with activated alumina, silica gel, or Drierite, and equipped with heating units (electrical or steam) for the regeneration of the drier. Normally, such units are provided in pairs, so that continuous operation may be assured. Fenwick[9] describes a unit which, along with others, is commercially available.

Under some circumstances in the field, where the equipment cannot be heated, anhydrous methanol (methyl alcohol) has been used to flush out moisture. The alcohol, having a lower boiling point, may be removed more readily than water with a stream of dry air, carbon dioxide or oil-pumped nitrogen, or with the application of a vacuum. Under no circumstances should methanol in quantity be allowed to remain in the system, since it will produce corrosion.[10] Methanol, in quantities somewhat less than 1%, does not induce rapid corrosion and moreover exerts an anti-freeze action.

Vacuum

5. Air is evacuated from equipment by using mechanical **vacuum pumps** or, in some instances, water suction pumps.[11] The latter are not in general desirable since fluctuation of water pressure may cause them to suck water back into the equipment being evacuated. Moreover, the vacuum obtained never exceeds the vapor pressure of the water used to operate the pump.

The boiling point of water is lower in a vacuum and consequently a lower temperature is required to evaporate moisture than at higher pressures. Water boils at approximately 104 F, under 28 in. of vacuum, and at 59 F, under 29.5 in. In general, temperatures much higher than these are employed with the vacuum method of drying.

Whether dry air or a vacuum is employed seems to make little difference, provided proper equipment is used for producing dry air or the desired vacuum, and the time and temperature requirements are met. Goddard[30] has shown that a vacuum treatment, followed by breaking the vacuum with dry air, when the vacuum period is concluded, gives additional drying, since the water remaining as a gas is removed by flushing with air. Carbon dioxide, oil-pumped nitrogen or gaseous refrigerant may be employed in lieu of air. Anderson[12] has shown one very desirable feature associated with the use of dry air, namely: when air is drawn through equipment restrictions and plug-ups are revealed; when a vacuum is employed neither is indicated. Blair and Calhoun[31] have pointed out a fallacy in the assumption that a low-pressure reading of an evacuated system insures the removal of all moisture in excess of that corresponding to this pressure. This is due to failure to reach true equilibrium conditions.

Little has been published concerning the **drying procedures** used by the major refrigerating machine manufacturers or employed by service engineers in the field. A few articles indicate the time, temperature, and moisture removal methods of a few procedures. The Westinghouse process for domestic units is described by Anderson[12] as involving a series of nine units, an oven temperature of 257 F, the use of air dried (not exceeding dew point of −58 F) with calcium chloride and potassium hydroxide and admitted under controlled low pressure, and a time (not revealed) sufficient for complete drying. A moisture test is made on the ninth unit.

Wile[13] describes, in very general terms, the drying method used by Savage Arms Corporation, for ice cream cabinet units. Better than ordinary drying, necessitated by the low temperatures involved, is attained by "dehydrating each piece of equipment . . . at high temperature and a very fine vacuum." This procedure apparently results in systems sufficiently dry, but silica gel drier units are placed on each machine as an additional precaution.

A general article[6] gives a drying procedure for sulfur dioxide machines, which includes heating in an oven at 250–275 F under vacuum (greater than 29.5 in.) for 4 hr. This is claimed to reduce the moisture content of the machine to approximately 0.0095 oz of water per cu ft of internal machine volume. As a means of removing the water vapor remaining in the machine, flushing with dry air is advocated. A discussion of drying by vacuum treatment at room temperature and flushing with dry air is also included in this article.

A number of articles have been written covering drying procedures employed by service men. Newcum[11] describes a method and apparatus for cleaning and dehydrating equipment in the field which involves drawing a vacuum on the system while heating it progressively toward the outlet connected to the vacuum pump. When this process is finished, dry hot air is admitted through a drier unit for some time. Properly carried out, this procedure should result in dry equipment. Fenwick[9] describes a system and apparatus for producing dry air and for drying units in the field as well as in the factory. Details of operation are given. A dehydrating unit which can be built in any well equipped shop is discussed by Dinsmoor.[14]

Control of Drying Operations

6. Proper **control** of drying operations is necessary if moisture difficulties are to be avoided. How may the manufacturer be certain that all equipment is dry and what routine checks may be used to catch "wet" machines before they reach the customer? Routine checking of the finished machine varies with the type of drying process employed. Where a stream of air is used, a dew-point determination of the outgoing air has been employed. Fenwick[9] advocates the use of a dew point of −40 F (equal to 0.046 gr water per cu ft) for the air leaving the equipment, as proof that it is dry.

Anderson[12] determines the residual moisture in the last machine of a series, which has been dried with air, by drawing off the moisture into a liquid air trap and from this calculating the quantity in the machine.

McGovern[5] recommends drawing a vacuum on the equipment being tested, closing off the system and observing the pressure rise, if any. If no pressure rise occurs after a time, the machine may be considered dry, since moisture remaining in the machine will create its own gas pressure.

Moisture may also be determined by removing a portion of the liquid refrigerant from a machine which has been operated for several hours and until the moisture is in equilibrium throughout. The quantity of moisture in the removed sample is determined by the phosphorous pentoxide method.[15] Recently Walker and Rinelli[16] developed a method applicable to methyl chloride, methylene chloride and the Freon refrigerants which makes possible routine analysis of a very large number of machines.

A standard[17] for refrigerant system tubing and cooling coil has been set at a maximum permissible moisture content of 10.6 milligrams per liter (300 milligrams per cu ft) of internal volume. A method for determining the moisture content was outlined.

Driers

7. After a machine has been installed, overhauled, recharged, etc., moisture is a very frequent cause of trouble. In the absence of other difficulties, a machine may be put into satisfactory operating condition by the removal of moisture with a drier. A drier is a chemical compound capable of adsorbing, or reacting chemically with the moisture contained in the liquid or gaseous refrigerant-oil mixture. The drier is placed in a suitable unit equipped with screens and filter pads to prevent the solid drier from entering the refrigerating system. The drier unit, in addition to holding the drier, functions as a filter and removes any solid found in the liquid or gaseous refrigerant reaching it.

There are several substances which have been used as driers in the chemical laboratory. Information concerning these driers has been published in a series of papers.[18,19,20] The data from these papers are not applicable directly to the problem of drying a refrigerant and consequently more is to be gained from several general articles.[21,22,23,6,24,3]

In order that a chemical substance be acceptable for use as a drier, it must possess, in addition to the property of water removal, characteristics which eliminate any possibility of undesirable reactions with the refrigerant, oil or machine parts.[6,24] Activated alumina, silica gel, Drierite, calcium oxide and calcium chloride have been most widely used, the first three being rated highly acceptable.

Table 3 contains the results of tests[3] and shows the relative efficiency of several driers.

Table 3. Drying Power of Various Materials

Drier	Refrigerant	Liquid or vapor	Maximum residual moisture,* % (Initial water concentration) .25%	.02%
Activated alumina	Sulfur dioxide	L	.15	.005
		V	.01	
	Methyl chloride	L	.02	.006
		V	.01	
Silica gel	Sulfur dioxide	L	.15	.006
		V	.01	
	Methyl chloride	L	.01	.004
		V	.01	
Drierite (Calcium sulfate)	Sulfur dioxide	L	.15	.009
		V	.08	
	Methyl chloride	L	.05	.005
		V	.04	
Calcium chloride $CaCl_2$	Sulfur dioxide	L	.09	.013
		V	.03	
	Methyl chloride	L	.10	.005
		V	.04	
Calcium oxide CaO	Sulfur dioxide	L	.20	—
		V	.15	
	Methyl chloride	L	.15	—
		V	.08	
Barium oxide BaO	Sulfur dioxide	L	.20	.017
		V	.15	
	Methyl chloride	L	.05	.006
		V	.05	
Zinc	Sulfur dioxide	L	.25	—
		V	.25	

* Excluding samples which were of definitely poor grade.

A brief discussion of the various driers which have been and/or are being used in the refrigeration industry is given below.

Activated alumina[25] is a granular aluminum oxide which removes moisture by adsorption. It also adsorbs acids.[24] It dusts slightly on handling but this is not a source of trouble. It has had extensive and satisfactory use with sulfur dioxide, methylene chloride, methyl chloride, Freon-11 and Freon-12, is used with sulfur dioxide in vapor phase (suction line) only, with other refrigerants in either suction or liquid lines, usually the latter. It may be left on the machine indefinitely.

Silica gel[26] is a glass-like silicon dioxide gel which removes moisture by adsorption; does not dust. (See additional remarks under activated alumina.) It also adsorbs acids.[27]

Drierite[28] is anhydrous calcium sulfate prepared as a granular white solid. It removes moisture by chemical action. It dusts somewhat more than activated alumina but this does not cause trouble. Drierite is also cast in sticks. (See additional remarks under activated alumina.)

Calcium oxide removes water and acid by chemical action. It dusts somewhat. It cannot be used with sulfur dioxide but is satisfactory for other refrigerants. It rates as a fair drier with tendency to powder when excess moisture is present.

Calcium chloride removes water by chemical action and is not rated as a drier capable of reducing the moisture content to a low level. It may be used with all refrigerants. Excess moisture produces a solution of calcium chloride which is highly corrosive. It will not remove acid.

Unlike activated alumina, silica gel, Drierite and calcium oxide, it should not be left on a machine longer than a few days, and calcium chloride should be used with caution.

Barium oxide removes moisture by chemical action but should be used with care since it has been known to cause explosions.[24] It powders considerably after reaction with water.

Magnesium perchlorate, barium perchlorate. These are powerful oxidizing agents and have caused serious explosions with oil and methyl chloride.

Phosphorus pentoxide is an excellent drier, but its fine powdery form makes it difficult to handle and produces extensive resistance to flow of gas and/or liquid. A mixture with quartz[29] has been accorded limited use.

Soda-lime is a mixture of calcium oxide and sodium hydroxide. Its use has been limited and it is not recommended as a satisfactory drier.

Factors in the Use of a Drier[3]

8. Since a drier is extremely sensitive to moisture, it must be protected from it at all times until ready for use. Handling a drier in moist air must be avoided, and for this reason it is recommended that the drier be obtained in a factory-packed unit, or that it be prepared for use in the unit under conditions which will preclude contamination by moisture.

The position of a drier unit on a machine is important. A liquid feed line should enter the drier unit at the bottom with the exhaust at the top. This arrangement will insure the most uniform contact between refrigerant and drier. If a drier is placed vertically in the suction line, the feed should be at the top with the exhaust at the bottom in order that the oil may be blown down through and out of the drier unit.

A very important factor bearing on the whole problem of the use of a drier is the question of rate of moisture removal and the time a drier unit must be left on the machine. A critical glance at the problem will show that the time is longer than is often assumed. To take a specific example, suppose that three drops of water get into an otherwise dry system containing 3 lb of refrigerant. These three drops of water would amount to about .02% moisture. Since this amount of water in methyl chloride is sufficient to cause discoloration of steel and may freeze out at temperatures slightly below zero, it would appear advisable to dry the system down at least to .01% moisture. Suppose now that ½ lb of methyl chloride circulates per hr, which would represent a typical normal load in a household machine. Then in the first hour not more than one-sixth of the moisture would be removed, even if the drier did a complete job of moisture removal. Furthermore, since the returning drier refrigerant would dilute the wet refrigerant, the feed liquid would become progressively drier and less and less water could be removed by the drier. Hence, under even these ideal conditions, considerably more than 6 hr would be required. In addition, moisture will tend to collect in the evaporator of a flooded system, because even in relatively small concentrations it vaporizes much less readily than the refrigerant.

Also, in either a flooded or dry type system, it is highly probable that at least part of the water will be dissolved or entrained in the oil entering the compressor and will take a long time to be carried over into the refrigerant cycle where it can be removed.

As a further consideration, it must be recognized that no drier will reduce the moisture content to zero even under ideal conditions, while under practical refrigeration conditions the efficiency of the drier is apt to be considerably less than ideal.

Acrolein is required in some cities as a warning agent for use with methyl chloride. Tests[8] show that activated alumina, silica gel and barium oxide reduce the acrolein content of methyl chloride approximately 80, 40 and 50%, respectively, over a period of 70 days. Calcium chloride, Drierite and calcium oxide do not appreciably reduce the acrolein content in the same time period.

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19. LOAD CALCULATIONS

TO SELECT refrigeration equipment which will result in a balanced system, it is first necessary to calculate the heat load of the refrigerator. A rational estimate requires an understanding of the heat sources and heat transfer. There are four general sources of heat in a refrigerator: wall heat gain, air changes, product load, miscellaneous. For a complete calculation, each of the four sources must be treated separately to determine the total.

1. **Wall heat gain.** The heat gain through walls, floor and ceiling will vary with the following factors: type of insulation, thickness of insulation, construction, outside wall area, and temperature difference between refrigerator and ambient air.

The heat transfer factor with cork or equivalent insulation at various temperature differences are listed in **Table 1.** The thickness of cork referred to in this table is actual cork or equivalent insulation thickness and not the overall wall thickness. The comparative values of various insulation materials can be found in Chapter 12, Table 4.

The coefficient of transmittance, U of the wall can be found by the following equation:

$$U = \frac{1}{1/f_i + L/k + 1/f_0} \qquad (1)$$

where

U = coefficient of transmittance, Btu/hr F sq ft

L = thickness of the wall, in.

k = conductivity of material in wall, Btu per hr F sq ft per in.

f_i = inside film or surface conductance, Btu/hr F sq ft

f_0 = outside film or surface conductance, Btu/hr F sq ft

f_i and f_0 are frequently used as 1.65 and 6 respectively.

With thick walls and low conductivity, the resistance L/k makes U so small that $1/f_i$ and $1/f_0$ have little effect, and can be omitted for calculation. As walls are seldom made of one material, the value L/k represents the composite value of the several materials used in series to the heat flow, and for a wall with flat parallel surfaces of materials 1, 2, 3, etc.,

$$U = \frac{1}{L_1/k_1 + L_2/k_2 + L_3/k_3 +} \qquad (2)$$

where L_1, L_2, L_3, etc. are the thicknesses and k_1, k_2, k_3, etc. are the conductivities of the several materials used.

Having established the coefficient of transmittance U, the heat leakage is given by the basic equation:

$$Q = UA\Delta t \qquad (3)$$

Table 1. Wall Heat Gain

(Btu per sq ft per 24 hr)

Insulation	Temperature difference (ambient temp minus refrigerator temp), deg F																
Cork or equivalent in.	1	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120
3	2.4	96	108	120	132	144	156	168	180	192	204	216	228	240	252	264	276	288
4	1.8	72	81	90	99	108	117	126	135	144	153	162	171	180	189	198	207	216
5	1.44	58	65	72	79	87	94	101	108	115	122	130	137	144	151	159	166	173
6	1.2	48	54	60	66	72	78	84	90	96	102	108	114	120	126	132	138	144
7	1.03	41	46	52	57	62	67	72	77	82	88	93	98	103	108	113	118	124
8	0.90	36	41	45	50	54	59	63	68	72	77	81	86	90	95	99	104	108
9	0.80	32	36	40	44	48	52	56	60	64	68	72	76	80	84	88	92	96
10	0.72	29	32	36	40	43	47	50	54	58	61	65	68	72	76	79	83	86
11	0.66	26	30	33	36	40	43	46	50	53	56	60	63	66	69	73	76	79
12	0.60	24	27	30	33	36	39	42	45	48	51	54	57	60	63	66	69	72
13	0.55	22	25	28	30	33	36	39	41	44	47	50	52	55	58	61	63	66
14	0.51	20	23	26	28	31	33	36	38	41	43	46	49	51	54	56	59	61
Single glass	27.0	1080	1220	1350	1490	1620	1760	1890	2030	2160	2290	2440	2560	2700	2840	2970	3100	3240
Double glass	11.0	440	500	550	610	660	715	770	825	880	936	990	1050	1100	1160	1210	1270	1320
Triple glass	7.0	280	320	350	390	420	454	490	525	560	595	630	665	700	740	770	810	840

Note: Where wood studs are used multiply the above values by 1.1

where

Q = heat leakage, Btu/hr
U = coefficient of transmittance, Btu/hr F sq ft
A = outside area of section, sq ft
Δt = difference between average outside temperature and average refrigerator temperature, deg F

The refrigeration industry has tended to standardize as to the minimum insulation of a refrigerator. **Table 2** is a list of recommended minimum insulation thicknesses.

Table 2. Minimum Insulation Thicknesses

Storage temperature F	Cork or equivalent thickness (Inches)	
	Northern U. S.	Southern U. S.
50 to 60	2	3
40 to 50	3	4
25 to 40	4	5
15 to 25	5	6
0 to 15	6	7
0 to −15	7	8
−15 to −40	9	10

The outside design temperatures for the major cities in the United States can be found in Outdoor Design Data **(Table 5)**.

If the refrigerator is exposed to the sun, additional heat will be added to the heat load. For practical purposes the temperature difference can be adjusted to compensate for the sun effect. The values given in **Table 3** are applicable throughout the 24 hours and are added to the ambient temperature in calculating wall heat gain.

2. Air changes. Each time the door of a storage room is opened some air from the outside enters the storage room. This warm outside air must be reduced to the temperature of the refrigerated space thus adding to the refrigeration load. It is difficult to determine the load with any degree of accuracy. The traffic in a refrigerator usually varies with its size or volume and the number of door openings is dependent upon the volume rather than the number of doors. The air changes listed in **Table 4** are based on experience and have proven practical.

Table 3. Allowance for Solar Radiation

(Degrees fahrenheit to be added to the normal temperature difference for heat leakage calculations to compensate for sun effect—not to be used for air conditioning design)

Type of surface	East wall	South wall	West wall	Flat roof
Dark colored surfaces such as Slate roofing Tar roofing Black paints	8	5	8	20
Medium colored surfaces, such as Unpainted wood Brick Red tile Dark cement Red, gray or green paint	6	4	6	15
Light colored surfaces, such as White stone Light colored cement White paint	4	2	4	9

Table 4. Average Air Changes per 24 hr for Storage Rooms due to Door Openings and Infiltration

Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr
200	44.0	6,000	6.5
300	34.5	8,000	5.5
400	29.5	10,000	4.9
500	26.0	15,000	3.9
600	23.0	20,000	3.5
800	20.0	25,000	3.0
1,000	17.5	30,000	2.7
1,500	14.0	40,000	2.3
2,000	12.0	50,000	2.0
3,000	9.5	75,000	1.6
4,000	8.2	100,000	1.4
5,000	7.2		

Note:
For heavy usage multiply the above values by 2
For long storage multiply the above values by 0.6

The heat removed to reduce one cubic foot of air from outside conditions to the refrigerator temperature is listed in **Table 6.** Inlet air temperature listed in Table 6 refers to the air which enters the refrigerator when the doors open. The heat gain, due to air changes, can be found from Tables 4 and 6.

Table 5. Outdoor Design Data

Location	Dry bulb temp, F	Wet bulb temp, F
Alabama		
Birmingham	95	78
Mobile	95	80
Arizona		
Phoenix	105	76
Tucson	105	72
Arkansas		
Fort Smith	95	76
Little Rock	95	78
California		
Bakersfield	105	70
Fresno	105	70
Los Angeles	90	70
Oakland	85	65
San Francisco	85	65
Colorado		
Denver	95	64
Connecticut		
Hartford	93	75
New Haven	95	75
District of Columbia		
Washington	95	78
Florida		
Jacksonville	95	78
Key West	98	78
Miami	91	79
Tampa	95	78
Georgia		
Atlanta	95	76
Savannah	95	78
Idaho		
Boise	95	65
Illinois		
Chicago	95	75
Peoria	96	76
Springfield	98	77
Indiana		
Evansville	95	78
Fort Wayne	95	75
Indianapolis	95	76
Terre Haute	95	78
Iowa		
Des Moines	95	78
Sioux City	78	78
Kansas		
Dodge City	100	78
Wichita	100	75
Kentucky		
Louisville	95	78
Louisiana		
New Orleans	95	80
Shreveport	100	78
Maine		
Eastport	90	70
Portland	90	73
Maryland		
Baltimore	95	78
Massachusetts		
Boston	92	75
Michigan		
Detroit	95	75
Lansing	95	75
Minneapolis		
Duluth	93	73
St. Paul	95	75
Mississippi		
Vicksburg	95	78
Missouri		
Kansas City	100	78
St. Louis	95	78
Montana		
Billings	90	66
Helena	95	67
Nebraska		
Lincoln	95	78
Nevada		
Reno	95	65
New Hampshire		
Concord	90	73
New Jersey		
Newark	95	75
Trenton	95	78
New Mexico		
Albuquerque	95	70
New York		
Albany	93	75
Buffalo	93	73
New York	95	75
North Carolina		
Charlotte	95	78
Wilmington	95	73
North Dakota		
Bismarck	95	73
Ohio		
Cleveland	95	75
Dayton	95	78
Oklahoma		
Oklahoma City	101	77
Oregon		
Portland	90	68
Pennsylvania		
Erie	93	75
Philadelphia	95	78
Pittsburgh	95	75
Rhode Island		
Providence	93	75
South Carolina		
Charleston	95	78
South Dakota		
Huron	95	75
Rapid City	95	70
Tennessee		
Chattanooga	95	76
Memphis	95	78
Nashville	95	78

Table 5. Outdoor Design Data (Concluded)

Location	Dry bulb temp, F	Wet bulb temp, F
Texas		
Amarillo	100	72
Corpus Christi	102	95
Dallas	100	78
El Paso	100	69
Houston	95	78
Utah		
Salt Lake City	95	65
Vermont		
Burlington	90	73
Virginia		
Norfolk	95	78
Richmond	95	78
Washington		
Seattle	85	65
Spokane	93	65
West Virginia		
Parkersbury	95	75
Wisconsin		
Green Bay	95	75
Madison	95	75
Milwaukee	95	75
Wyoming		
Cheyenne	95	85

Table 4 does not apply if additional ventilation is required. When fresh air is provided, it causes the refrigerated space to be under an air pressure slightly above atmospheric. Under these conditions, cold air will leave the refrigerated space whenever the door is opened. The ventilating load under these conditions will replace the door opening load, if greater. To calculate the load due to ventilation, change the ventilating cubic feet per minute to cubic feet per 24 hours (cfm×60×24) and use Table 6 for the heat load per cubic foot of air.

3. Product load. A product placed in a refrigerator at a temperature higher than the storage temperature will lose heat until it reaches the storage temperature. The quantity of heat to be removed may be calculated from a knowledge of the product, including its state upon entering the refrigerator, its final state, its weight, specific heat above and below freezing, its freezing temperature, and latent heat. When a definite weight of product is cooled from one state and temperature to another state and temperature, some or all of the following calculations must be made:

Heat removal from initial temperature to some lower temperature above freezing:

$$Q = W \times c \times (t_1 - t_2) \qquad (4)$$

Heat removal from initial temperature to freezing point of product:

$$Q = W \times c \times (t_1 - t_f) \qquad (5)$$

Heat removal to freeze product:

$$Q = W \times h_{if} \qquad (6)$$

Heat removal from freezing point to final temperature below freezing:

$$Q = W \times c_i \times (t_f - t_3) \qquad (7)$$

where

Q = heat removed, Btu
W = weight of product, lb
c = specific heat of product above freezing, Btu/lb
t_1 = initial temperature above freezing, F
t_2 = lower temperature above freezing, F
t_f = freezing temperature of product, F
h_{if} = latent heat of fusion, Btu/lb
c_i = specific heat of product below freezing, Btu/lb
t_3 = final temperature below freezing, F

The product load is represented by the sum of the several loads given above. If it is required to remove the heat from the product in a given number of hours the following formula may be used to evaluate the equivalent load based on 24 hour calculations:

$$\frac{\text{Btu}}{\text{24 hr}} = \frac{\text{Product load} \times 24}{\text{Hr reqd for product load}}$$

Specific heats above and below freezing and **latent heats** of **fusion** for many products are given in Table 5, Chap. 12. It should be noted that the latent heat of fusion has a definite relationship to the water content of the product; in the absence of data on the latent heat of a product and with a knowledge of its moisture content, the latent heat may be estimated by multiplying the percentage of water expressed as a decimal by 144, the latent heat of fu-

Table 6. Heat Removed in Cooling Air to Refrigerator Conditions
(Btu per cu ft)

Storage room temp F	Temperature of outside air, F							
	85		90		95		100	
	Relative humidity, %							
	50	60	50	60	50	60	50	60
65	0.65	0.85	0.93	1.17	1.24	1.54	1.58	1.95
60	0.85	1.03	1.13	1.37	1.44	1.74	1.78	2.15
55	1.12	1.34	1.41	1.66	1.72	2.01	2.06	2.44
50	1.32	1.54	1.62	1.87	1.93	2.22	2.28	2.65
45	1.50	1.73	1.80	2.06	2.12	2.42	2.47	2.85
40	1.69	1.92	2.00	2.26	2.31	2.62	2.67	3.06
35	1.86	2.09	2.17	2.43	2.49	2.79	2.85	3.24
30	2.00	2.24	2.26	2.53	2.64	2.94	2.95	3.35

Storage room temp F	Temperature of outside air, F							
	40		50		90		100	
	Relative humidity, %							
	70	80	70	80	50	60	50	60
30	0.24	0.29	0.58	0.66	2.26	2.53	2.95	3.35
25	0.41	0.45	0.75	0.83	2.44	2.71	3.14	3.54
20	0.56	0.61	0.91	0.99	2.62	2.90	3.33	3.73
15	0.71	0.75	1.06	1.14	2.80	3.07	3.51	3.92
10	0.85	0.89	1.19	1.27	2.93	3.20	3.64	4.04
5	0.98	1.03	1.34	1.42	3.12	3.40	3.84	4.27
0	1.12	1.17	1.48	1.56	3.28	3.56	4.01	4.43
−5	1.23	1.28	1.59	1.67	3.41	3.69	4.15	4.57
−10	1.35	1.41	1.73	1.81	3.56	3.85	4.31	4.74
−15	1.50	1.53	1.85	1.92	3.67	3.96	4.42	4.86
−20	1.63	1.68	2.01	2.09	3.88	4.18	4.66	5.10
−25	1.77	1.80	2.12	2.21	4.00	4.30	4.78	5.21
−30	1.90	1.95	2.29	2.38	4.21	4.51	4.90	5.44

sion of water. Most food products have a freezing temperature in the range of 26 to 31 F, with an average freezing temperature of about 28 F. If the exact freezing temperature is unknown, it may be assumed to be 28 F. See Applications Volume, ASRE Data Book (Sec. I), for a discussion of food freezing.

To illustrate the foregoing methods, assume that 100 lb of lean beef are to be cooled from 90 to 40 F, after which it is to be frozen and cooled to 0 F.

From Table 5, Chap. 12, the specific heat of lean beef before freezing is 0.77 Btu/lb and after freezing is 0.40 Btu/lb. The latent heat of fusion is 100 Btu/lb.

To cool from 90 F to 40 F in chill room:

$$100 \times 0.77 \times (90-40) = 3\,850 \text{ Btu}$$

To cool from 40 F to freezing point in freezer:

$$100 \times 0.77 \times (40-28) = 924 \text{ Btu}$$

To freeze:

$$100 \times 100 = 10{,}000 \text{ Btu}$$

To cool from freezing point to storage temperature:

$$100 \times 0.40 \times (28-0) = 1120 \text{ Btu}$$

Total 15,894 Btu

Note that in this example the latent heat of fusion represents approximately two-thirds of the total product load. In calculations involving freezing of a product the latent heat of fusion will always be a high percentage of the product load.

Table 7. Heat Equivalent of Electric Motors

Motor hp	Btu/hp hr		
	Connected load in refr space[1]	Motor losses outside refr space[2]	Connected load outside refr space[3]
$\frac{1}{8}$ to $\frac{1}{2}$	4,250	2,545	1,700
$\frac{1}{2}$ to 3	3,700	2,545	1,150
3 to 20	2,950	2,545	400

[1] For use when both useful output and motor losses are dissipated within refrigerated space; motors driving fans for forced circulation unit coolers.

[2] For use when motor losses are dissipated outside refrigerated space and useful work of motor is expended within refrigerated space; pump on a circulating brine or chilled water system, fan motor outside refrigerated space driving fan circulating air within refrigerated space.

[3] For use when motor heat losses are dissipated within refrigerated space and useful work expended outside of refrigerated space; motor in refrigerated space driving pump or fan located outside of space.

Fresh fruits and vegetables give up heat during storage. It must be remembered that they are alive and continue to undergo changes during storage. The most important of these changes is produced by respiration, a process in which the oxygen of the air is combined with the carbon of the plant tissue. During this process, energy is released in the form of heat. The amount of heat liberated varies with the type and temperature of the product. The colder the product, the less the heat of respiration. The rate of evolution of heat of various products is listed in **Table 9.**

4. Miscellaneous. All electrical energy dissipated in the refrigerated space (lights, motors, heaters, etc.) must be included in the heat load. Each watt is equal to 3.42 Btu per hr. Heat equivalents of electric motors are shown in **Table 7.** People give up heat at varying rates depending upon the temperature, type of work, clothing, size, etc. The average hourly load due to occupancy is shown in **Table 8.** When people go into the refrigerated space for short durations, they will carry with them a considerable amount of heat over and above that listed in Table 8. Therefore, some allowance must be made if the traffic load of this type is heavy.

Short method. There are many short or rule-of-thumb methods of estimating load. Some are based on external surface only. This is basically incorrect since the usage load will depend upon the volume and the volume of a refrigerator is not proportional to the external surface. To illustrate, assume two refrigerators with both external areas equal to 1600 sq ft. If they are both 10 ft high, one may have floor dimensions of 35×10 ft, another 20×20 ft. Calculation will show a 12.5% variation in volume.

A short method based on volume as well as surface will give more dependable results than a method based on surface only. Of the four general sources of heat in the refrigerator: wall losses, air change, product, and miscellaneous, the external surface will always be proportional to the wall losses. The problem of simplification is with the remaining three sources in terms of volume. **Table 10** gives the probable values for the gains due to air changes, product, and miscellaneous. The values are based on experience and judgment must be used in its application.

To find the hourly load, the total 24 hour load is divided by the desired compressor operating time. When a natural defrost cycle by means of heat from the refrigerator air is desired (35 F and above), it has been general practice to calculate the load based on 16 hour operation. When the refrigerant temperature is 30 F or higher, a defrost cycle is not required, and general practice has been to select equipment based on 18 or 20 hour operation. For refrigerator temperatures below 34 F some other method of defrosting must be provided. All of these methods add some heat to the refrigerator. The amounts of heat vary considerably with different methods. It is suggested that the recommendations of the manufacturer be followed on the heat added due to defrosting and the sug-

Table 8. Heat Equivalent of Occupancy

Cooler temperature F	Heat equivalent/person Btu/hr
50	720
40	840
30	950
20	1,050
10	1,200
0	1,300
−10	1,400

Table 9. Approximate Rate of Evolution of Heat by Certain Fresh Fruits and Vegetables when Stored at the Temperatures Indicated*

Commodity	Temperature	Heat evolved per ton of fruits or vegetables per 24 hours*
	F	British thermal units
Apples	32	660 to 1,000
	40	1,110 to 1,760
	60	4,400 to 6,600
	85	6,600 to 15,400
Bananas:	54	3,300
Green	68	8,360
Turning	68	9,240
Ripe	67	8,360
Beets	32	2,650
	40	4,060
	60	7,240
Cantaloupes	32	1,320
	40	1,960
	60	8,500
Carrots	32	2,130
	40	3,470
	60	8,080
Celery	32	2,820
	40	4,540
	60	13,520
Cherries (sour)	32	1,320 to 1,760
	60	11,000 to 13,200
Grapefruit	32	460
	40	1,070
	60	2,770
	80	4,180
Grapes: Cornichon and Flame Tokay	36	660 to 1,100
	60	2,200 to 2,640
	80	5,500 to 6,600
Sultanina	32	430
	43	1,050
	53	1,690
Emperor	32	350
	43	850
	53	1,810
Ohanez	32	300
	43	740
	53	1,570
Lemons	32	580
	40	810
	60	2,970
	80	6,200
Lettuce	32	11,320
	40	15,990
	60	45,980

Commodity	Temperature	Heat evolved per ton of fruits or vegetables per 24 hours*
	F	British thermal units
Mushrooms (cultivated)	32	6,160
	50	22,000
	70	58,000
Onions (Yellow Globe)	32	660 to 1,100
	50	1,760 to 1,980
	70	3,080 to 4,180
Oranges	32	690 to 900
	40	1,400
	60	5,000
	80	8,000
Peaches	32	850 to 1,370
	40	1,440 to 2,030
	60	7,260 to 9,310
	80	17,930 to 22,460
Pears (Bartlett)	32	660 to 880
	60	8,800 to 13,200
Peppers	32	2,720
	40	4,700
	60	8,470
Potatoes (Irish Cobbler)	32	440 to 880
	40	1,100 to 1,760
	70	2,200 to 3,520
Raspberries	36	4,400 to 6,600
	60	15,400 to 17,600
Strawberries	32	2,730 to 3,800
	40	5,130 to 6,600
	60	15,640 to 19,140
	80	37,220 to 46,440
String beans	32	4,740
	40	6,740
	60	22,630
Sweet corn	32	5,890
	40	8,190
	60	17,130
Sweet potatoes Not cured	32	2,440
	40	3,350
	60	6,300
Cured	32	1,190
	40	1,710
	60	4,280
Tomatoes (mature green)	32	580
	40	1,070
	60	6,230
Tomatoes (ripe)	32	1,020
	40	1,260
	60	5,640

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* The figures in this table supersede those in Table 1, Circ. 278, U. S. Department of Agriculture, edition of November, 1941.

Table 10. Usage Heat Gain, Btu per 24 hr for One cu ft Interior Capacity

Volume cu ft	Service*	Temperature difference (ambient temp minus refrigerator temp), deg F										
		1	40	50	55	60	65	70	75	80	90	100
20	Average	4.68	187	234	258	281	305	328	351	374	421	468
	Heavy	5.51	220	276	303	331	358	386	413	441	496	551
30	Average	2.88	115	144	159	173	188	202	216	230	259	288
	Heavy	4.56	182	228	251	274	297	319	342	365	410	456
50	Average	2.28	91.2	114	126	137	148	160	171	182	205	228
	Heavy	3.55	142	177	196	213	231	249	267	284	320	355
75	Average	1.85	74.0	92.5	102	111	120	130	139	148	167	185
	Heavy	2.88	115	144	158	173	188	202	216	230	259	288
100	Average	1.61	64.3	80.5	88.5	96.6	105	113	121	129	145	161
	Heavy	2.52	101	126	139	151	164	176	189	202	227	252
200	Average	1.38	55.2	69.0	75.9	82.8	89.7	96.6	103	110	124	138
	Heavy	2.22	88.8	111	122	133	144	155	166	178	200	222
300	Average	1.30	52.0	65.0	71.5	78.0	84.5	91.0	97.5	104	117	130
	Heavy	2.08	83.2	104	114	125	135	146	156	166	187	208
400	Average	1.24	49.6	62.0	68.2	74.4	80.6	86.8	93.0	99.2	112	124
	Heavy	1.96	78.4	98.0	108	118	128	137	147	157	176	196
500	Average	1.21	48.4	60.5	66.6	72.6	78.7	84.7	90.7	96.8	109	121
	Heavy	1.87	74.8	93.5	103	112	122	131	140	150	168	187
600	Average	1.17	46.8	58.5	64.4	70.2	76.0	81.9	87.7	93.6	105	117
	Heavy	1.85	74.0	92.5	102	111	120	130	139	148	167	185
800	Average	1.11	44.4	55.5	61.1	66.6	72.2	77.7	83.3	88.8	100	111
	Heavy	1.76	70.4	88.0	96.8	106	115	123	132	141	158	176
1000	Average	1.10	44.0	55.0	60.5	66.0	71.5	77.0	82.5	88.0	99.0	110
	Heavy	1.67	66.8	83.5	91.9	100	108	117	125	134	150	167
1200	Average	.995	39.8	49.8	54.7	59.7	64.7	69.7	74.7	79.6	89.6	99.5
	Heavy	1.58	63.2	79.0	86.9	94.8	103	111	119	126	142	158
1500	Average	.920	36.8	46.0	50.6	55.2	59.8	64.4	69.0	73.6	82.8	92.0
	Heavy	1.50	60.0	75.0	82.5	90.0	97.5	105	113	120	135	150
2000	Average	.835	33.4	41.8	45.9	50.1	54.3	58.5	62.7	66.8	75.2	83.5
	Long storage	.775	31.0	38.8	42.6	46.5	50.4	54.3	58.1	62.0	69.8	77.5
3000	Average	.750	30.0	37.5	41.3	45.0	48.8	52.5	56.2	60.0	67.5	75.0
	Long storage	.576	23.0	28.8	31.7	34.6	37.3	40.3	43.2	46.1	51.8	57.6
5000	Long storage	.403	16.1	20.2	22.2	24.2	26.2	28.2	30.2	32.2	36.3	40.3
7500	Long storage	.305	12.2	15.3	16.8	18.3	19.8	21.4	22.9	24.4	27.5	30.5
10,000	Long storage	.240	9.60	12.0	13.2	14.4	15.6	16.8	18.0	19.2	21.6	24.0
20,000	Long storage	.187	7.48	9.35	10.3	11.2	12.2	13.1	14.0	15.0	16.8	18.7
50,000	Long storage	.178	7.12	8.90	9.79	10.7	11.6	12.5	13.4	14.2	16.0	17.8
75,000	Long storage	.176	7.04	8.80	9.68	10.6	11.5	12.3	13.2	14.1	15.8	17.6
100,000	Long storage	.173	6.92	8.65	9.52	10.4	11.2	12.1	13.0	13.8	15.6	17.3

* For average and heavy service, product load is based on product entering at 10 deg above the refrigerator temperature; for long storage the entering temperature is approximately equal to the refrigerator temperature.

Where the product load is unusual do not use this table, see pages 338-342.

gested operating time. It has been general practice to calculate the load for refrigerator temperature below 34 F based on 18 hour operation.

The calculated hourly load is used as a guide in selecting equipment. Since the compressor sizes are limited to comparatively large increments, the compressor selection must be guided by the calculated load, and the actual selection of compressor may change the calculated time up or down. The selection of the evaporator should be made in balance with the selected compressor capacity, and not with the original calculated load.

Example 1. Given: Refrigerator 20×10×10 ft, 4 in. cork insulation.

90 F ambient temperature
60% relative humidity
30 F refrigerator temperature
2,000 lb of beef at 40 F to be cooled to 30 F in 24 hr
200 watts electrical load

Solution:

1000 sq ft outside surface
90 − 30 = 60 F temperature reduction
(1) Wall gain
1000×108 (Table 1) = 108,000 Btu per 24 hr
(2) Air change load
19×9×9 = 1540 cu ft
10.3 air changes per 24 hr (Table 4)
2.53 Btu per cu ft to cool air from 90 F, 60% relative humidity to 30 F (Table 6)
1540×10.3×2.79 = 44,200 Btu per 24 hr
(3) Product load
Specific heat = .77 (Table 5, Chap. 12)
2000×.77×(40 − 30) = 15,400 Btu per 24 hr
(4) Miscellaneous (electrical load)
1 watt = 82 Btu per 24 hr
200×82 = 16,400 Btu per 24 hr
(5) Total load

Wall gain	108,000
Air change	44,200

Product	15,400
Electrical	15,400
	184,000
Add 10% safety factor	18,400
	202,400 Btu per 24 hr

202,400 ÷ 16 = 12,560 Btu per hr for 16 hr compressor operation

Using Short Method (Table 10)

(1) Wall gain same as above 108,000 Btu per 24 hours

(2) Usage—average 60 F temperature difference 55.2 Btu per cu ft per 24 hr
1,540 × 55.2 = 85,200 Btu per 24 hr
108,000 + 85,200 = 193,200 Btu per 24 hr
193,200 ÷ 16 = 12,080 Btu per hr for 16 hr compressor operation.

Example 2. Given: Refrigerator, 15 × 11 × 9 ft, 4 in. cork insulation, two triple glass windows each 6 × 5 ft

90 F ambient temperature
40 F refrigerator temperature
800 watt lights and motors from 8 a.m. to 4 p.m.
Two men working from 8 a.m. to 4 p.m.
2000 lb beef to be cooled from 60 F to 40 F from 6 p.m. to 6 a.m.

Solution:

60 sq ft glass surface
738 sq ft net insulated surface
90 − 40 = 50 F temperature reduction

(1) Wall gain
738 × 90 (Table 1) = 66,400 Btu per 24 hr
60 × 350 (Table 1) = 21,000 Btu per 24 hr
66,400 + 21,000 = 87,400 Btu per 24 hr

(2) Air change load
14 × 10 × 8 = 1120 cu ft
16.6 air changes per 24 hr (Table 4)
2.26 Btu per cu ft to cool air from 90 F to 40 F (Table 6)
1,120 × 16.6 × 2.26 = 42,000 Btu per 24 hr

(3) Product load
Specific heat of beef is .77 (Table 5, Chap. 12)

$$\frac{2000 \times .77 \times 20 \times 24}{12} = 61{,}600 \text{ Btu per hr}$$

(4) Miscellaneous
Men working
2 × 750 = 1500 Btu per hr
Electrical
800 × 3.42 = 2736 Btu per hr

Comparison Day and Night Loads

Btu per 24 hr (A) Night Load		Btu per 24 hr (B) Day Load
87,400	Wall gain	87,400
42,000	Air change	42,000
61,600	Product	
	Men	36,000
	Electrical	65,600
191,000		231,000
19,100	10% Safety factor	23,100
210,100	Btu per 24 hr	254,100

Total B is larger than Total A and should be used as the load.

254,100 ÷ 16 = 15,900 Btu per hr for 16 hr compressor operation.

Example 3. Given: Banana storage room 15 × 10 × 8 ft outside, 3 in. cork insulation

95 F ambient temperature
65 F storage room temperature
Storage 8000 lb ripe bananas coming in at 65 F
300 watts electrical load

Solution:

700 sq ft outside surface
95 − 65 = 30 F temperature reduction

(1) Wall gain
700 × 30 × 2.4 (Table 1) = 50,400 Btu per 24 hr

(2) Air change load
14 × 9 × 7 = 882 cu ft
18.8 air changes per 24 hr (Table 4)
1.54 Btu to cool 1 cu ft from 95 F to 65 F (Table 6)
882 × 18.8 × 1.54 = 25,500 Btu per 24 hr

(3) Product load
8360 Btu per 24 hr per ton, heat of respiration (Table 9)

$$\frac{8000}{2000} \times 8360 = 33{,}400 \text{ Btu per 24 hr}$$

(4) Electrical load
300 × 3.42 × 24 = 24,600 Btu per 24 hr

(5) Total load

Wall gain	50,400
Air change	25,500
Product	33,400
Electrical	24,600
	134,000
Add 10% safety factor	13,400
	147,400 Btu per 24 hr

147,400 ÷ 18 = 8200 Btu per hr for 18 hr compressor operation

When the refrigerant temperature is above 30 F ice will not form on the coil, therefore a defrost cycle is not necessary.

20. MOTORS AND ELECTRICAL CONTROLS

THE principal applications in the refrigerating industry requiring electric motors are reciprocating compressors, fans and centrifugal compressors. The types of motors required for these applications depend on a number of factors such as the torque requirements of the driven device, the degree of unloading during the starting operation, and the electrical power supply available. Consideration of the proper motor to do the job also determines the type of enclosure required for the motor to protect it against severe atmospheric conditions.

Fractional hp motors. There are two general classifications of motors in the industry—the fractional hp motors, and the integral hp motors. Fractional hp motors comprise ratings smaller than 1 hp at 1,800 rpm, and are of a wide variety of types and sizes depending on the application. Most motors in this classification are single-phase, although occasionally polyphase and dc motors are employed.

The various types of motors and their usual application are as follows:

Type	*Maximum hp*	*Applications*
Split-phase	$\frac{1}{3}$	Small compressors & fans
Capacitor start-induction run	$\frac{3}{4}$	Compressors & fans
Repulsion induction	$\frac{3}{4}$	Compressors & fans
Capacitor start & run	$\frac{3}{4}$	Overhung fans—unit heaters & coolers (not suitable for belted application)
Shaded pole	$\frac{1}{10}$	Overhung fans—unit heaters & coolers (not suited for belted loads)
Direct current	$\frac{3}{4}$	Compressors & fans

Integral hp motors. In the integral horsepower classification, single-phase motors are employed on many applications, usually up to a maximum of 5 hp, but occasionally up to 10 hp in those cases where polyphase power is not available. The various types of motors and their applications in the refrigerating industry are as follows:

Type	*hp*	*Applications*
Single-phase—capacitor start-induction run	1 to 5	Compressors & fans
Single-phase—repulsion induction	1 to 5	Compressors & fans
Squirrel-cage—normal torque 2 or 3-phase	1 & up	Compressors started unloaded and fans
Squirrel-cage—high torque 2 or 3-phase	1 & up	Compressors started loaded
Slip ring—2 or 3-phase	1 & up	Fans, centrifugal compressors and reciprocating compressors
Synchronous—2 or 3-phase	50 & up	Centrifugal compressors and reciprocating compressors unloaded

Single-Phase Motors

Split-phase motors. The split-phase motor depends upon the principle of a stator having two windings, one displaced with respect to the other, causing a magnetic drag on the conductors of a squirrel-cage motor, in order to produce the starting operation. One of the windings in the stator, called the running winding, is made of relatively coarse wire, and the other winding, or starting winding, is made of very fine wire. These two windings are connected in parallel across the single-phase line when starting. The starting winding by reason of its fine wire has high resistance, and, therefore, higher power factor than the main winding, which causes the phase displacement between the two windings necessary to produce a rotating magnetic field. When a predetermined speed is reached, approximately 70% of synchronous speed, a centrifugal device on the squirrel-cage rotor opens a switch in series with the starting winding, and thereafter the motor operates as a single-phase, squirrel-cage induction motor, drawing current only through the running winding.

These motors are available in various ratings from ⅙ hp up to and including ½ hp, and with either rigid or resilient mounting, depending upon the requirements of the application. They are suitable for application either direct-connected or with belt drives to small fans and blowers.

These motors are provided either with four leads, or with a terminal block, in order to permit reversing the direction of rotation. To reverse the direction of rotation, it is merely necessary to reverse the two leads to the starting winding. A typical speed torque curve of this type of motor is shown in **Fig. 1.**

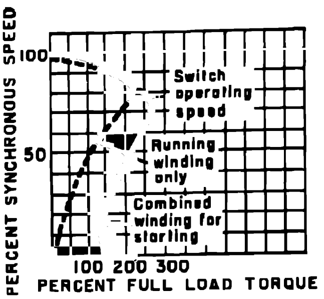

Fig. 1. Speed Torque Curves of General-Purpose Split-Phase Motors

Another type of split-phase motor, called the high-torque, split-phase motor, is sometimes used in refrigerating and air conditioning work, although it is not recommended for such applications on account of the fact that the starting currents for this type of motor are high enough to permit possibility of lamp flicker. For this reason this high-torque motor normally used on washing machines, ironers, and home work-shop machines, is not used in refrigeration work requiring fairly frequent starting.

The general-purpose split-phase motor, and all other types of fractional hp motors described in the following paragraphs, have locked rotor currents within the following NEMA Standards for 115-volt motors, which standards are approved by practically all power companies.

⅙ hp and below —20 amperes
¼ hp —23 amperes
⅓ hp —31 amperes
½ hp —45 amperes
¾ hp —61 amperes

The minimum efficiencies, power factors and apparent efficiencies of 60-cycle, general-purpose motors of fractional hp ratings, are shown in **Table 1.** These figures are, in most cases, conservative compared with the performance of motors which may be obtained on the market. The starting currents for motors on 230-volt lines are half the above figures.

Capacitor start-induction run motors. This type of motor is available not only in fractional hp ratings, but also in integral horsepower ratings up to a maximum of 5 hp. In principle they are fundamentally the same as the split-phase type of motor, except that the starting winding is provided with a capacitor in series with it, and in this manner it is possible to obtain high starting torques without exceeding the

Table 1. Minimum Efficiencies, Power Factor, and Apparent Efficiencies of 60-Cycle General-Purpose Motors, Percentage

(NEMA Standards, MG-18-101, July, 1941)

Horsepower	Efficiency				Power factor				Apparent efficiency			
rpm	3,600	1,800	1,200	900	3,600	1,800	1,200	900	3,600	1,800	1,200	900
poles	2	4	6	8	2	4	6	8	2	4	6	8
⅛	45	53	45	38	57	52	43	36	28	30	21	15
⅙	49	58	49	42	62	56	46	38	34	36	25	18
¼	53	62	53	45	66	60	49	40	39	42	29	20
⅓	54	63	54	46	67	61	50	41	41	44	31	22
½	55	65	55	47	69	63	52	43	44	47	33	23
¾	57	67	57	49	72	65	53	44	46	49	34	24

Note: Apparent efficiency = (Efficiency × power factor)/100.

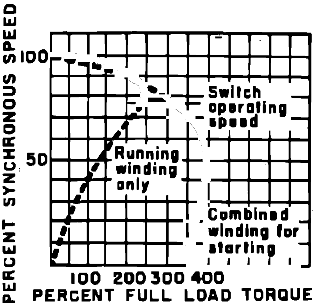

Fig. 2. Speed Torque Curves of Capacitor Start-Induction Run Motors

starting current limitations mentioned in the above tabulation. A typical speed torque curve for this type of motor is shown in **Fig. 2.**

The capacitor start-induction run motors, because of their high starting torque and low starting current, are ideal for compressors, refrigeration and air conditioning equipment. Typical performance specifications for this type of motor are shown in **Table 2**.

The integral hp motors of the same type have speed torque curves fundamentally the same as for the fractional hp rating, except that the various torque values are slightly lower, in order to hold the starting current to a minimum. They will have a starting torque of 300 to 325%, a torque at transfer of 225 to 250%, and a pull-out torque of at least 200%.

There is such a wide variety of possible connections for capacitor type motors that these connections will not be covered. The nameplate on the motor usually gives sufficient information for making the proper connections.

Capacitor start and run motors. In this type of motor there are two windings in the stator, the same as in the capacitor start-induction run motor, except that both windings are left in the circuit at all times. A capacitor is provided in series with the second winding, and the value of the capacitor is selected to give the best operating condition with the motor at full speed. For this reason, this type of motor has an exceptionally low starting torque and therefore is not recommended for belted applications. Its principal field of application is that of unit heaters and unit coolers where the fan is mounted directly on the motor shaft. In these cases the motor is usually totally enclosed, depending for its cooling on the air being drawn over the motor by the fan. Its principal advantage over other types of motors is that it does not require the use of any centrifugal device for cutting out a starting winding. This type of motor lends itself readily to speed control down to 50% of full-load speed.

Shaded pole motors. Very small fans are often driven by small shaded pole motors. These motors, also single-phase, are available in sizes up to about $\frac{1}{10}$ hp. In this type of motor, the stator is provided with a single winding consisting of one coil around each salient pole, and the coils connected in series across the line. In order to produce the rotating magnetic field necessary to produce starting torque, each pole piece is provided with a shading coil which is a single turn of copper surrounding only half of the pole iron. In operation, the al-

Table 2. Typical Specifications for Fractional hp Capacitor Type Refrigeration Compressor Motors

(Values shown are representative for 115-volt, 60-cycle, 4-pole, single-phase motors)

	Horsepower rating				
	$\frac{1}{6}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$	$\frac{3}{4}$
Full load current amperes	3.0	3.7	4.6	7.0	10.0
Locked-rotor (starting) current, amperes	16.2	22.0	25.0	40.0	58.0
Full load efficiency, percentage	63.0	68.0	70.0	72.0	73.0
Full load power factor, percentage	57.0	62.0	66.0	64.0	65.0
Full load torque, oz ft	8.1	12.2	16.2	24.4	36.5
Break-down (maximum) torque, oz ft	25.0	34.0	45.0	71.0	106.0
Locked-rotor (starting) torque, oz ft	40.0	57.0	69.0	109.0	155.0

ternating current in the main winding sets up flux in the field core 90 electrical degrees out of phase with the current. This flux by transformer action induces current in the shading coil, and the current in the shading coil sets up flux in the half of the iron which it surrounds. This type of motor has relatively high slip. The rotor is a simple squirrel-cage type and no centrifugal switch mechanism is employed.

Very low starting torques are developed with this type of construction, being of the order of 40 to 50% of full load torque so that the application of the motor is limited to direct-connected fans only, and should never be belted. Its efficiency and power factor are exceptionally low, but these factors are not of much importance in motors of such small size. The physical location of the shading pole determines the direction of rotation, and the rotation of the motor cannot be changed in the field except by dismantling the motor and bringing the shaft out the opposite end of the motor, providing, of course, the construction permits.

Repulsion start-induction run motors. This type of single-phase motor also finds a very wide field of application in the refrigeration and air conditioning industry, particularly for belt drive to small size compressors not only in the fractional horsepower sizes, but also for integral horse power ratings up to 5 hp. Occasionally they will be used up to 10 hp.

A straight single-phase repulsion motor has a distributed winding in the stator, while the rotor is similar to a direct-current armature with windings connected to a commutator. When the brushes are all short-circuited by connecting them together, and are set in the correct position, this motor is capable of developing a very high starting torque. However, instead of having a constant speed characteristic so essential to compressor drive, it has a drooping speed torque curve. To take care of this condition, the repulsion start-induction run motor is provided with a centrifugal device that connects all the commutator bars together at a predetermined speed, thereby converting the armature to a squirrel-cage rotor.

Thus this motor combines the high starting torque of the repulsion motor with the constant speed characteristic of the squirrel-cage motor while running. Some motors of this type are so designed that the brushes are automatically lifted from the commutator immediately after the bars have been connected. This feature eliminates brush noise and wear while running.

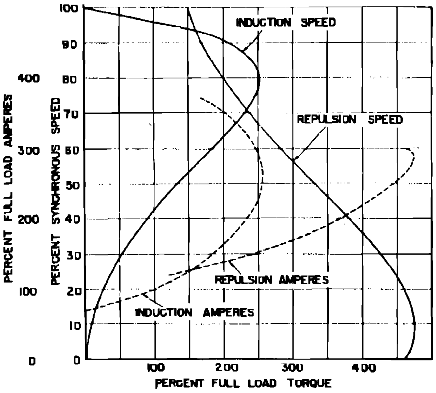

Fig. 3. Speed Torque and Current Torque Curves of Typical Repulsion Start-Induction Run Single-Phase Motor

Fig. 3 shows the typical performance of the repulsion start-induction run motor. The motor torque follows the repulsion torque curve up to the point where the centrifugal device short circuits the commutator. Thereafter it follows the induction curve. The optimum speed for short-circuiting the bars occurs when the repulsion and induction curves cross.

Table 3 shows minimum torque values for fractional hp repulsion start-induction run motors as established by NEMA.

The repulsion-induction motor is similar in construction, but without any device for short-circuiting the commutator or lifting the brushes. Instead, it has a squirrel-cage winding deeply embedded in the rotor underneath the repulsion winding. As a result, the speed torque curve is a modification of the combined torque curves of both windings over the entire speed range.

These two types of motors have the advantage that they develop more starting torque per ampere than any other type of single-phase motor. However, because of commutator and brush maintenance, high

Table 3. Minimum Torque Values of Repulsion Start-Induction Motors, in Percentage of Full Load Running Torque

(NEMA Standards MG8-104, July, 1941)

hp	Rating 60 cycles, 1,725 rpm			Rating 60 cycles, 1,140 rpm		
	Locked rotor	Pull-up	Breakdown	Locked rotor	Pull-up	Breakdown
$\frac{1}{8}$	350	200	200	300	150	185
$\frac{1}{6}$	350	200	200	300	150	185
$\frac{3}{4}$	350	200	200	300	150	185
$\frac{1}{3}$	350	200	200	300	150	185
$\frac{1}{2}$	350	200	200	300	150	185
$\frac{3}{4}$	350	200	200	—	—	—

manufacturing costs and radio interference during starting, they are gradually being replaced by the capacitor start-induction run type motor.

Polyphase Induction Motors

Polyphase induction motors are the simplest and most rugged, and because of these features are more universally used in industry than any other type. They have a wide variety of characteristics, depending on the application, and are available over the complete range of horsepowers from the smallest fractional horsepower up to the largest integral ratings. They can be obtained for either two or three-phase systems for operation on 110, 208, 220, 440, or 550-volt lines, and in the larger sizes for operation on 2,300 volts or even higher. Most systems in the United States are 60-cycle, but motors can also be obtained for 25, 40 or 50 cycles where the power systems are of these frequencies.

For the purpose of describing the principle of operation the two-phase is probably the more simple. It has two sets of stator windings located in the motor 90 electrical degrees apart, and respectively connected across the two phases of the line. The phase displacement between the currents in the two windings produces a rotating magnetic field, causing a drag on the conductors of the rotor, thus enabling it to start and come up to full speed. The rotor is usually made up of laminations with a simple uninsulated bar type winding in it. This type is called the **squirrel-cage motor.** By varying the shape, size and resistance of these bars, the torque characteristics can be varied to meet the particular application for which the motor is intended. The rotors can also be of the slip ring or wound rotor type.

Squirrel-cage type motors are divided by NEMA into several classes giving a wide variety of speed torque characteristics. These classifications are:

Class A—Normal starting torque with normal inrush current
Class B—Normal starting torque with low inrush current
Class C—High starting torque with low inrush current
Class D—High slip motors.

NEMA Standards MG-4-35 have assigned a letter designation to appear on nameplates of all motors ¼ hp and larger to indicate the locked rotor KVA per hp input to the motor. This standard is shown in **Table 4.** The locked rotor KVA with respect to the locked rotor starting current bears the following relationships:

$$\text{KVA} = \frac{\sqrt{3} \times \text{volts} \times \text{locked amperes}}{1{,}000} \qquad \text{For 3-phase motors}$$

$$\text{KVA} = \frac{2 \times \text{volts} \times \text{locked amperes}}{1{,}000} \qquad \text{For 2-phase motors}$$

$$\text{KVA} = \frac{1 \times \text{volts} \times \text{locked amperes}}{1{,}000} \qquad \text{For single-phase motors}$$

Class A normal torque-normal inrush motors. This type of motor, usually called the general-purpose motor, is rapidly being superseded by the Class B motor. It may have starting currents as high as eight times or even ten times full load current, and for this reason it is not looked upon with favor, even though its efficiency and power factor may be very slightly better than those of the Class B motor.

It can be applied to fans, blowers, centrifugal pumps and to reciprocating compressors when started unloaded.

Class B normal torque-low inrush motors. The normal torque-low inrush motor is more widely used in the refrigerating and air conditioning industry than any other type. It is successfully applied to fans, blowers, centrifugal pumps and to reciprocating compressors when started unloaded.

The locked rotor or starting torques and the maximum running torques of this class of motors are shown for different pole combinations in **Table 5.**

Fig. 4 shows a typical speed torque curve for this type of motor as well as the Class C high torque motor.

In **Fig. 5** are shown full load efficiencies and power factors for the average motors of different horsepowers and speeds from 1 up to 400 hp. In the average motor, the efficiencies for ½ and ¾ load will be 1 and 2 or 2 and 3%, respectively, below the figures on the curves for full load. A typical efficiency and power factor curve is shown in **Fig. 6.**

Table 4. Letter Code Designation Showing Starting KVA per hp for Alternating Current Motors ½ hp and Larger

Letter designation	KVA per hp
A	0 - 3.14
B	3.15- 3.54
C	3.55- 3.99
D	4.0 - 4.49
E	4.5 - 4.99
F	5.0 - 5.59
G	5.6 - 6.29
H	6.3 - 7.09
J	7.1 - 7.99
K	8.0 - 8.99
L	9.0 - 9.99
M	10.0 -11.19
N	11.2 -12.49
P	12.5 -13.99
R	14.0 -and up

Note: Motors manufactured prior to the date of issuance of this Code do not include the KVA per hp on the nameplate.

Table 5. Locked Rotor and Maximum Running Torque Values for NEMA Class B Squirrel Cage Motors

No. of poles	Min locked rotor torque, %	Max running torque, %
2	150	200
4	150	200
6	135	200
8	125	200
10	120	200
12	115	200
14	110	200

Note: The locked rotor torque and maximum running torque for Class B motors will not be less than the values shown in table above except that some 2-pole designs have lower locked rotor torques than 150%.

Class C high torque-low inrush motors. By means of a rotor having special shapes of bars in the squirrel-cage winding, high starting torques in the neighborhood of 200 to 250% of full load torque are developed without a material sacrifice in running efficiency and power factor. This type of motor, somewhat more expensive than the Class B, is called the Class C high torque-low inrush motor. Its typical speed torque curve is shown in Fig. 4 along with the curve for the Class B motor.

The principal field of application for this motor is for starting reciprocating compressors which are required to be started up with load.

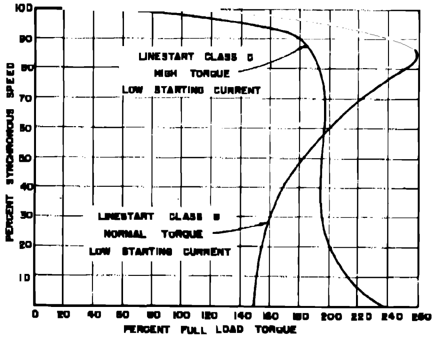

Fig. 4. Speed Torque Curves, Class B and C Motors

Class D high slip motors. This motor is seldom used in the refrigerating and air conditioning industry.

Multi-speed squirrel-cage motors. On many fan applications and compressors it is desirable to have more than one speed and to obtain it as simply and inexpensively as possible. Two-speed, three-speed and four-speed motors in many cases will fulfill the requirements even though the speed control is not so close as desired. Where only two speeds are required with a ratio of two to one in speed, this can be accomplished with one winding in the motor, reconnected externally by means of a drum switch or contactors to give the second speed. The connections for a two-speed variable or constant torque motor such as would be required for a fan drive are shown in **Fig. 7.**

Where a speed range as high as two to one is not desired, the stator must be provided with two separate windings and inter-locked in the control, so that only one can be energized at any one time. In other words, the motor functions as two separate motors in one housing.

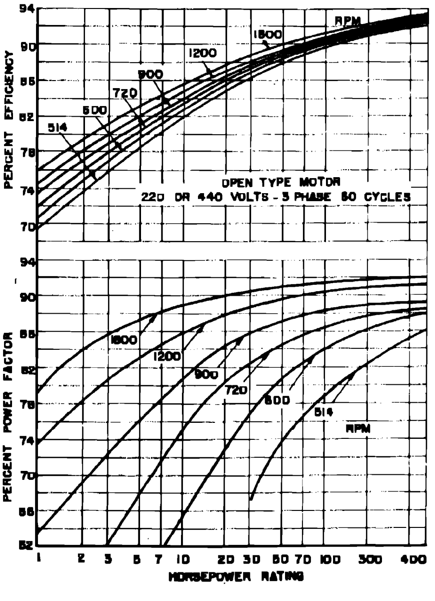

Fig. 5. Representative Full Load Efficiencies and Power Factors of NEMA Class B Squirrel Cage Motors

If three or four speeds are desired, they can be obtained with two stator windings, each one a duplicate of the consequent pole winding shown in Fig. 7.

Wound rotor (slip ring). In those cases where a speed range of two to one is required with quite a number of operating speeds between these limits, or where the motor is of such a large size that the power company system cannot stand the starting current of a squirrel-cage motor, the wound rotor (slip ring) motor is often employed.

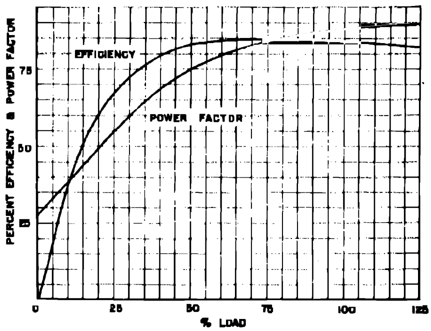

Fig. 6. Motor Efficiencies and Power Factors for Motors at Part Load

This motor is similar in construction to the squirrel-cage motor as far as the stator is concerned, but the rotor, instead of being uninsulated, is provided with the same insulated type of winding and the three leads connected to collector rings. Carbon brushes on the rings are connected to an external resistor and drum controller or contactors to cut the resistance out of the circuit with each successive step of the control. With sufficient resistance in the external rotor circuit the initial starting torque, and likewise the initial starting current, can be reduced to full load values or even lower.

Fig. 8 shows a typical set of speed torque curves of a wound rotor motor with a seven point control driving a fan or centrifugal compressor, where the torque required by the fan varies as the square of the speed. On the first point of the controller, it will be observed that the motor will develop approximately 70% starting torque with approximately 75 to 80% of full load current. Since this torque is in excess of the break-

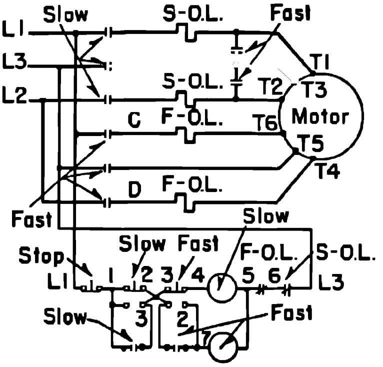

Fig. 7. Schematic Connection Diagram for Two-Speed, Single Winding Variable or Constant Torque, Squirrel-Cage Motor with Magnetic Control

away torque requirements of the fan, the motor will start to accelerate the load and eventually bring the fan up to 55% speed. Then, if the controller is moved over to the second point, speed torque curve, R-2 applies and the motor will accelerate still further up to 65%. By further increase in the controller, the motor and fan will be brought up to the final or top speed without causing excessively high currents to be drawn from the line. If the operator does not remain on point one of the controller, but moves rapidly over to point two, the initial starting torque will be 125% of full load torque, and the initial starting current likewise of the same proportion.

This type of motor, particularly in the larger sizes, is admirably suited to driving centrifugal compressors through step-up gearing. Where reduced refrigerating capacity is required on moderately warm days in air conditioning systems, it is more economical from a power consumption standpoint to operate the compressor at reduced speeds rather than run it at full speed and vary the output of the compressor by damper control.

Hermetic Motors

According to NEMA, "A hermetic motor consists of a stator and rotor without shaft, and shields or bearings for installation in refrigeration condensing units of the hermetically sealed type."

In other words, with the motor and compressor sealed in a common chamber, its

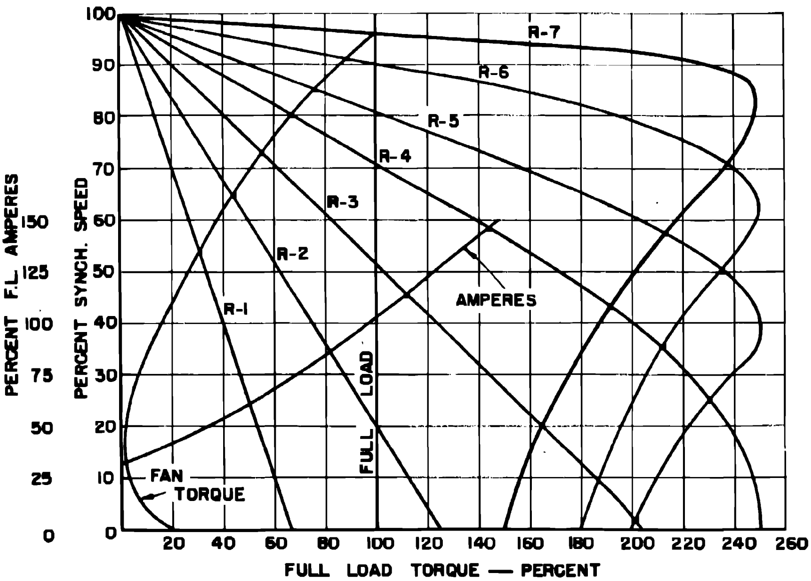

Fig. 8. Speed Torque Curves of Wound Rotor Motor

construction must be such that its insulation can withstand the action of the refrigerant and lubricating oil. Special selected long-fiber cotton treated with special compounds and a few types of resin-coated wires are acceptable to the application in that they neither foul the refrigerant and lubricant, nor deteriorate due to their action.

Hermetic motors are available through the fractional horsepower range in the split-phase, capacitor start and two-value capacitor single-phase types of construction. In these cases, starting switches and auxiliaries are mounted external to the compressor unit.

Integral horsepower ratings are available for single-phase power, but by far the highest percentage of installations above 3 hp are 3-phase. Hermetically sealed units are available up to 100 hp.

Temperature rise. Unlike the normal open-type motor, the hermetic motor can be cooled by a variety of means such as air, water, refrigerant vapor, refrigerant liquid, oil or a combination of these means.

The cooling of an hermetic motor may vary with compressor operating conditions as much as with load, so that the temperature rise may serve as a limitation under one set of compressor operating conditions and the maximum torque requirements under another set.

Inasmuch as the temperature rise of a hermetic motor depends as much upon the design of the compressor and cooling system as upon the motor losses, the motor manufacturer ordinarily does not guarantee the temperature rise of a hermetic motor, and the compressor manufacturer is ultimately responsible for the temperature rise.

Torque values of hermetic motors. In the fractional hp type, hermetic motors are available in low, normal and high starting torque designs. In the normal and high torque designs, the starting torques are in the range of 300 to 400% of full load torque and the break-down torque approximately 250%.

In the integral hp polyphase type, the starting torques will vary from 250 to 300% of full load torque with the break-down torques having the same values.

Locked rotor currents or starting currents of hermetic motors usually do not exceed the values recommended by NEMA for complete induction motors of similar rating.

Synchronous Motors

Synchronous motors, because of their higher efficiency and ability to correct power factor, have a very definite field in the refrigerating and air conditioning industry, particularly in the larger sizes, 100 hp and above. Their principal fields of application are in driving slow-speed reciprocating compressors and in furnishing power through step-up gearing to centrifugal compressors used in air conditioning.

Synchronous motors differ from induction motors in that the rotating field receives its excitation from a source of direct current, either in the form of a direct-connected exciter, or from direct current bus excitation. Usually in the air conditioning and refrigeration field, they are used on 220, 440, or 2,300-volt systems. Whereas induction motors always operate with a lagging power factor, the synchronous motor operates at unity power factor or a leading power factor. Leading power factor machines are usually designed for 80% where a considerable amount of correction is required. Both the 100% and 80% machines, when operating at reduced load, furnish additional power factor correction in accordance with **Fig. 9.**

The unity power factor synchronous motors are usually 1 to 2% higher in efficiency than the corresponding rating of induction motor either of the squirrel-cage or wound rotor types. The 80% power factor synchronous motor, because it has a higher full load current and, consequently, higher losses than the 100% power factor machine, has just about the same efficiency as the induction motor of the same horsepower rating.

Slow-speed synchronous motors. Slow-speed synchronous motors of the engine type construction (that is, without shaft or bearings) are usually employed to drive slow-speed reciprocating compressors. These motors have speeds of 450 rpm and lower. Because of the fact that such compressors are always started up unloaded, the starting torque and pull-in torque requirements are quite low, usually on the order of 30 to 50% of full load torque. Because of these low torque requirements for starting, the current inrush is usually in the neighborhood of 250 to 275% of full load current for the unity power factor motors, and 225 to 250% of full load current for the 80% machines. With these low values of starting inrush, this type of motor is invariably started across full line voltage.

High-speed synchronous motors. The high-speed synchronous motor, also available for unity power factor or 80% leading power factor operation, finds its principal application in driving high-speed centrifugal compressors through step-up gearing. As mentioned previously, the wound rotor type of motor is more economical from the power consumption standpoint where load variation is required, but there are many installations involving more than one compressor where the base refrigeration load can be handled by the synchronous motor type of drive, and the variations in load taken care of by the slip ring motor on another compressor.

There is also another form of centrifugal compressor drive utilizing the synchronous motor where varying load requirements are encountered, and that is the drive utilizing a variable speed fluid coupling between the synchronous motor and the step-up gearing. The overall efficiency of this drive is just about comparable with the slip-ring motor application. It works out more advantageously in those cases where space is at a premium because of the relatively large space requirements of the secondary resistor for the wound rotor application.

High-speed synchronous motors inherently have higher starting currents than the low-speed type of motor, so that even though the torque requirements for starting the centrifugal compressor are quite low the inrush on full voltage is in the neighborhood of 500% of full load current. Some power systems can satisfactorily handle this inrush, while on others it is necessary to resort to reduced voltage starting. In applications of this nature, it is imperative that the particular power company involved be consulted regarding starting current limitations. The motor and control manufacturer should be provided with complete information as to the horsepower requirements, speed and WR^2 of the compressor, together with start-current limitations and interrupting capacity of the power system involved.

Direct-Current Motors

In some cases where alternating current is not available, direct-current motors are used to drive the various fans, blowers, pumps, and compressors required by this industry. This type of motor, available over the complete range from the smallest fractional horsepower up to the largest integral horsepower rating, will always have ample torque available for acceleration for running loads within its rating. Motors are rated either 115 or 230 volts dc.

Fractional horsepower motors, except the extremely small ratings, are all compound wound, and can be thrown directly across the line. Integral horsepower motors can be furnished either as shunt wound or compound wound machines. If compound wound, they can be thrown across the line up to 3 hp, and for voltages 230 volts and below. Larger compound wound motors, and all integral horsepower, shunt wound

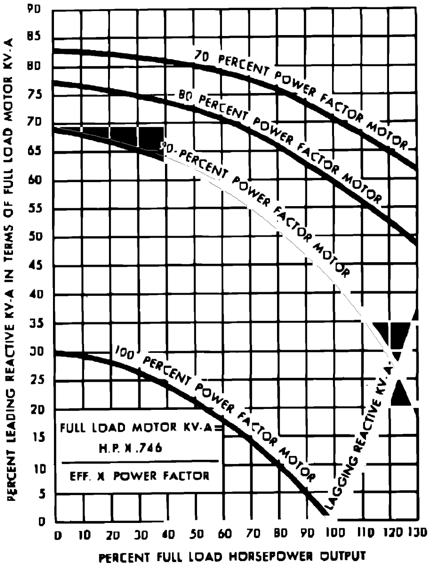

Fig. 9. Leading Reactive KVA Obtainable with Synchronous Motors under Various Load Conditions

motors require a starting resistance in series with the armature.

One advantage of the shunt wound dc motor over any of the ac motors is the fact that by shunt field control, a wide range of operating speed, with a large number of operating points and with efficient operation, can be obtained. For this reason, shunt wound motors are often used for driving fans.

Reciprocating compressors, especially if required to start against load and to start and stop frequently, should use the compound wound type motor in order to be able to accelerate with the minimum amount of current drawn from the line.

For Induction and Synchronous Motors

$$\text{Synchronous speed} = \frac{120 \times f}{P}$$

where f = frequency in cycles/sec

P = number of poles

$$\%\ \text{slip} = \frac{\text{synchronous speed} - \text{FL speed}}{\text{FL speed}}\ .$$

Types of motor enclosures. All of the various types of motors heretofore described can be obtained with a wide variety of mechanical enclosures to protect them from extreme operating conditions. The various types of enclosures normally used in this industry, together with their temperature rise guarantees, are as follows:

Open-type motor	40 C rise continuous
Drip-proof motor	50 C rise continuous
Splash-proof motor	50 C rise continuous
Totally enclosed motor*	55 C rise continuous
Totally enclosed fan-cooled motor*	55 C rise continuous

* Including explosion-proof motors.

Ambient temperatures. All of the above motors are guaranteed to deliver their full horsepower continuously under operating conditions where the ambient temperature does not exceed 40 C. If higher ambient temperatures are to be encountered, special motors should be applied having a temperature rise lower by the amount that the actual ambient exceeds 40 C.

Service factor. All standard open-type motors carry a service factor of 1.15 which means that they will carry 15% overload continuously without reaching dangerous temperatures, providing both the voltage and the frequency are held constant at the nameplate values.

Drip-proof, splash-proof, totally en-

General Information on Motors

Useful Formulae

$$\text{FL amps} = \frac{\text{hp} \times 746}{\text{Eff.} \times \text{P.F.} \times \sqrt{3} \times E} \qquad \text{for 3-phase motors}$$

$$\text{FL amps} = \frac{\text{hp} \times 746}{\text{Eff.} \times \text{P.F.} \times 2 \times E} \qquad \text{for 2-phase motors}$$

$$\text{FL amps} = \frac{\text{hp} \times 746}{\text{Eff.} \times \text{P.F.} \times E} \qquad \text{for single-phase motors}$$

$$\text{KVA input} = \frac{E \times I \times \sqrt{3}}{1{,}000} \qquad \text{for 3-phase motors}$$

$$\text{KVA input} = \frac{E \times I \times 2}{1{,}000} \qquad \text{for 2-phase motors}$$

$$\text{KVA input} = \frac{E \times I}{1{,}000} \qquad \text{for single-phase motors}$$

$$\text{Kw input} = \text{KVA input} \times \text{P.F.}$$

$$\text{Hp output} = \frac{\text{kw input} \times \text{eff}}{.746}$$

$$\text{Hp output} = \frac{\text{torque} \times \text{rpm}}{5{,}250}$$

closed, and totally enclosed fan-cooled motors do not carry this service factor.

Insulation. There are three different classes of insulation used in electric motors. Class A insulation, the kind used in most motors, involves the use of paper, cotton and resinous-coated conductors with similar materials for slot-cell insulation and phase insulation with the entire winding covered or impregnated with an insulating varnish. This class of insulation is good for a maximum hot spot temperature of 105 C.

Class B insulation includes asbestos, mica and glass, together with normal insulating varnish, and is good for a maximum hot spot temperature of 135 C.

Class C insulation includes glass and ceramics for which no maximum temperature limitation has been set. In the present state of the art, this type of insulation is not being supplied for standard motors.

Thermal protection for motors. In many motors, especially of the single-phase type, thermal protective devices are sometimes incorporated in the motor construction. They consist of bimetal discs or strips in intimate contact with the motor iron or windings, and incorporating in their construction contacts which open the circuit in case the motor for any reason reaches a dangerous operating temperature. In the single-phase motors, there is included in this device a heater carrying the motor line current so as to give adequate protection against stalled motor conditions.

In the polyphase type motor, the same type of protection is provided except that the contacts in the device open the control circuit, and since such motors carry currents of appreciable magnitude, no auxiliary heaters are included. This type of thermostatic protection will not protect this motor against stalled rotor conditions, but will protect it against overheating caused by overloads, too frequent starting or lack of proper ventilation.

Effect of voltage variations on motor characteristics. For the following types of ac motors—split-phase, capacitor type, repulsion start-induction run type, or the polyphase induction motor—all torques vary as the square of the voltage.

For example, a repulsion start-induction run motor attempting to start on 90% voltage would have its speed torque curve as shown in Fig. 3 reduced to 81% of the values shown in the curve. In other words, the starting torque would be reduced from 460% to 370%, the torque at transfer from repulsion to induction operation reduced from 235% to 190%, and the maximum torque reduced from 250% to 200%. The operating speed would be decreased a very slight amount, the efficiency would be decreased several per cent, but the power factor would be increased. The average motor would operate at full load with a slightly higher temperature rise. Conversely, if the voltage were 10% high, the torque values would increase 21% above the figures shown in Fig. 3.

In the case of the synchronous motors, the starting torques and pull-in torques vary as the square of the voltage, but the pull-out torque or maximum torque varies directly with the voltage.

All motors are guaranteed to operate satisfactorily and deliver their full rated horsepower at rated frequency under voltage conditions 10% above or below nameplate rating without reaching dangerous operating temperatures.

Effect of frequency variations on motor characteristics. All motors will operate satisfactorily under conditions of plus or minus 5% frequency variation, providing the voltage is held constant. They will also operate satisfactorily where the sum of the voltage and frequency variation does not exceed 10% (provided the variation in frequency does not exceed 5%) above or below normal rating as stamped on the motor nameplate.

The question often arises as to whether or not a 60-cycle motor can be used on 50 cycles. Usually standard open-type polyphase motors will operate satisfactorily, delivering their rated load with a 50 C temperature rise, instead of 40 C rise.

Single-phase, 60-cycle motors are not usually adapted to operation on a 50-cycle circuit on account of the fact that the speed at which the centrifugal device operates will be too high.

Adequacy of power supply. Because of the harmful effect which low-voltage conditions have on the speed torque curves and operating temperatures, it is imperative that all transformers and feeders serv-

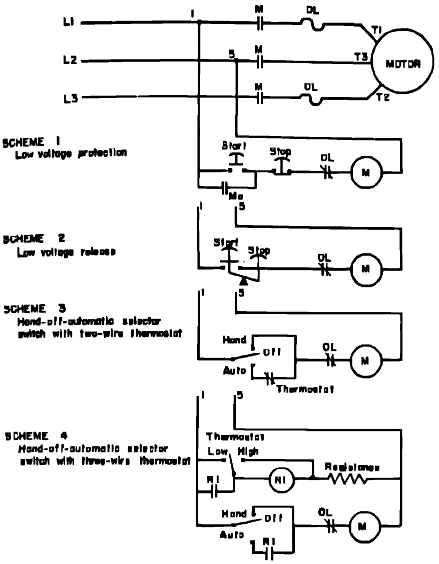

Fig. 10. Schematic Diagram of Across-the-Line Magnetic Starter

ing air conditioning and refrigerating loads be of adequate size to take care of normal running loads, and also starting currents without causing too much voltage drop. Many refrigerating and air conditioning motors are applied on the basis of a duty cycle loading, so that under certain conditions of operation, they are required to start and come up to speed against a very material overload. It is important, therefore, that voltages be held up to normal at all times. Many installations have experienced serious difficulties because of inadequate transformer and feeder capacities.

Electrical Controls for Motors

Functions of control. The principal functions of any control device are as follows:

1. Start and stop the motor either manually, magnetically by means of push buttons or magnetically by means of controlling devices such as thermostats, pressure relays, etc.
2. Protection to the operator
3. Protection to the motor
4. Control the speed
5. Limit the current drawn from the line, during starting or during running.

Manual control is provided for many single-phase motors and polyphase motors up to and including 3 hp. This control consists of a manually operated set of line contacts, provided with a set of thermal overload relays which will trip the contacts to the open position in case a sustained overload occurs.

Other manually controlled devices include a wide variety of rheostats and speed regulators for obtaining limited speed control on capacitor motors used on unit heater and unit cooler drives.

Across-the-line magnetic starters. This type of starter, more widely used than any other on polyphase induction motors, is adaptable to either manual starting and stopping by means of push-buttons, or it can be automatically controlled by means of thermostats or pressure relays. **Fig. 10** shows a schematic diagram of this starter, with push button control or with selector switch and thermostat control.

Where push buttons alone are employed, they may be wired up for either low-voltage release or low-voltage protection, both of which are shown in Fig. 10. Low-voltage protection, which means protection to the operator, utilizes a three-wire start-stop pushbutton station so arranged that in case of voltage failure the motor will not start up again on resumption of voltage until the start pushbutton is pressed. Low-voltage release involves the use of a two-wire walking beam type of pushbutton whereby, in case of power failure, the motor will stop but will start up again automatically upon resumption of voltage.

Auto transformer starters. The auto transformer type of starter is used on larger squirrel-cage motors above 20 or 25 hp where it is necessary to limit the starting current drawn from the line. A schematic diagram of this type of starter is shown in **Fig. 11**.

With this type of starter the current drawn from the line varies as the square of the voltage applied across the motor terminals. It has the disadvantage that open transition takes place upon changing from starting to running, and for this reason is sometimes considered objectionable by the power company on whose lines it is operating.

Resistance type (increment type). This

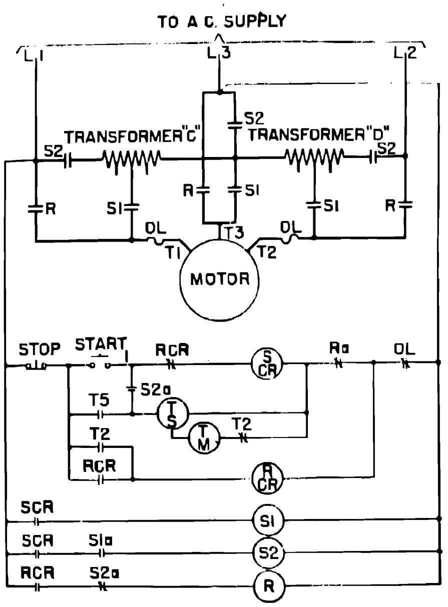

Fig. 11. Typical Schematic Diagram of Magnetic Auto-Transformer Type Starter for Squirrel Motors

type of starter for the squirrel-cage induction motor is often used on the 208-volt network systems in the larger cities. It employs a line contactor, one or more steps of resistance in the leads going to the motor, and one or more accelerating contactors to short out this resistance in bringing the motor up to speed. The advantage of this type of starter is that it does not open the circuit to the motor at any time during the starting operation, but it has the disadvantage that the current drawn from the line is directly proportional to the voltage across the motor terminals. Where two or more steps of resistance are employed, as a result of power company limitations, these limitations may be such that the motor may not start on the first step of the control, which condition should not cause any undue alarm.

The power company regulations may be such that the full locked rotor current will not cause any disturbance on the line, providing this value of current is reached by increments occurring not oftener than once every one and a half or two seconds. A schematic diagram for this type of starter is shown in **Fig. 12.**

Wound rotor motor control. This type of control usually employs a combination of an across-the-line starter in the primary circuit to the motor in combination with a manually-operated or motor-operated drum controller and resistance controlling the secondary winding of the motor to give the desired speed control.

Synchronous motor starters. Synchronous motor starters may be of the full voltage or reduced voltage type as required. If of the reduced voltage type they may incorporate either resistance, reactance or auto-transformer type starting. Also, they may be either of the semi-magnetic or full magnetic type.

In the semi-magnetic type, manually operated breakers are used to apply reduced voltage or full voltage to the motor stator windings, after which at the desired time or speed direct current is automatically applied to the field. In the full magnetic type, the entire operation is automatically handled in the correct sequence merely by pressing a start pushbutton.

Various sequence and extra protective relays, not normally furnished with the starters used for squirrel-cage motors, are required in the synchronous motor starter.

Maintenance of Electrical Equipment

When a new motor is first received on the job, a thorough **inspection** should be made to insure that it has not been damaged in shipment, and that it is in good working condition. If possible, the shaft should be turned by hand to see that it rotates freely.

Bearings should be inspected before power is applied to see that they are provided with adequate lubricant. Fractional hp motors are normally provided with waste packed sleeve bearings, and are usually supplied with enough oil to last three or four months in operation. An inspection of the bearings will reveal whether or not oil should be added. This type of motor should use a light grade of good machine oil (S.A.E. #10 W).

If the motor is provided with ring oiled sleeve bearings, oil must be added to the proper level before it is placed in service.

If the motors are provided with ball bearings, they will have sufficient lubricant for at least three or four months' service. Some motors are provided with pre-lubricated sealed ball bearings which will only require cleaning and re-packing once every five years.

Cleanliness is of utmost importance in the satisfactory operation of any motor. Periodically, depending upon the amount of dirt in the atmosphere, motor windings should be thoroughly cleaned by blowing them out with air, using care that the blast of air from a compressed air hose does not come too close to the motor windings and cause damage to the insulation. Every three to five years it is advisable to clean the motor windings thoroughly and give them another coat of insulating varnish.

When new motors are received, they should be stored in a warm, dry place until ready to be used. If, for any reason, they have been exposed to undue moisture conditions, they should be thoroughly dried out before full voltage is applied to them.

The most effective method for **drying** the larger higher voltage motors is to pass current through the windings, using a voltage low enough to be safe when the windings are moist. For 2,300-volt motors, 220 volts will usually be satisfactory for circulating drying out current. Thermometers should be placed on the windings to see that they are being heated uniformly, and that the temperature does not exceed 90 C. For medium size and lower voltage motors, drying out may be accomplished by enclosing the machine in a temporary housing of sheet metal, and placing space heaters about the machine.

Where **brushes** are employed on direct-current, slip-ring or synchronous motors, the brushes should be inspected periodically to see that they are free in the holders, and that they are not worn excessively. The brush riding surfaces should be inspected, and if it is found that they are picking up metal from the rings or from the commutator, the metal particles should be removed, and the brush face resurfaced. Brush tensions on the average machine should be on the order of 2 to 2.5 psi of projected brush area.

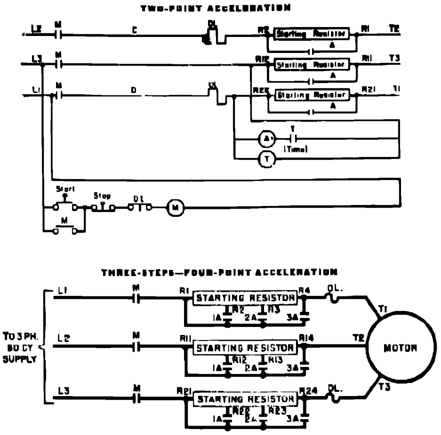

Fig. 12. Schematic Wiring Diagrams of Resistance Type Magnetic Reduced Voltage Starters for Squirrel-Cage Induction Motors

Air gaps in the average induction motor vary from 0.012 in. in the smaller motors up to 0.038 in. in the larger ratings. It is important that the air gaps be checked periodically to insure that bearing wear has not permitted the rotor to strike the stator. When the air gap on one side of the motor reaches a point where it is half or two-thirds that on the other side, it is an indication that the bearings are probably worn, and that they should be replaced.

All **control apparatus** should be inspected thoroughly each time the motor is given an inspection. If contacts are badly pitted they should be dressed up, and if badly worn they should be replaced. All control connections should be checked up periodically to see that they are tight. Most control contactors do not require any lubrication and should, therefore, not be oiled.

21. COOLING TOWERS

Nomenclature

A = the basic loading area; the gross cross sectional face area of a counterflow, forced convection cooling tower, sq ft
a_e = the effective composite area per unit of active filled volume, sq ft per cu ft
a_f = the area of active filling surface per unit of active filled volume, sq ft per cu ft
B = barometric pressure, in. Hg
b = the active width of atmospheric tower, ft
C_z = cycle modulus gpm per sq ft per cycle
c = the active tower length of atmospheric tower center to center of end posts, ft
d = dynamic splash effect, sq ft per gpm
ε = filling splash height per cycle, in.
F = wet-bulb factor
f_c = the overall coefficient of heat exchange by conduction and convection, Btu per sq ft, F min
f_d = the vaporization coefficient for mass transfer of water vapor, lb per sq ft min, and in lb per lb
GPM_b = basic loading gpm per sq ft = $0.120L_2/A$ or $0.120\ L_2/bc$.
H = the vertical distance center to center of splash decks in an atmospheric cooling tower, in.
H_f = height of fill in a packed tower or height of active spray chamber in a spray filled tower, ft
I = sizing index, Figures 11 through 15.
L_2 = the initial water rate, lb per min
M = a subscript indicating a mean effective value
m = loading factor, sq ft per gpm per deck
N = the number of splash decks in an atmospheric cooling tower
P = wetted perimeter of filling element, in.
r = deck factor, sq ft per gpm per deck
S = sizing scale abscissa of Figure 6, decks
ΔS = sizing scale intercept, decks
s = the humid specific heat of the air-water mixture ($s = 0.24 + 0.45w$), Btu per lb dry air F
t_1 = the final water temperature, F
t_2 = the initial water temperature, F
v = gross vertical linear velocity of air in a forced convection tower, against filling face, ft per min
v_{max} = maximum linear air velocity over filling, ft per min
V_a = the humid volume of air entering the tower, cu ft per lb dry air
W = wind velocity factor, Table 1
w = the humidity ratio (specific humidity), lb water vapor per lb dry air
z = the cycle height, in. per cycle
$\simeq$ = sensible equivalence
Σ_t = the sigma function for saturated air at the temperature of the bulk of the liquid, Btu per lb dry air
Σ = the sigma function for saturated air at the temperature of the main air stream, Btu per lb dry air
α = percentage of filling surface wetted at C_z

1. The refrigerating machine is operated for the purpose of extracting heat from spaces which are maintained at a temperature lower than that of the natural surroundings. The useful heat load on the stream of the refrigerant plus the energy added by the compressor must be discarded through a suitable heat exchanging apparatus. The first step in this discard is the transfer of the heat through the condenser walls to a receiving fluid—water or air. Where natural waters for condensers are not available, or where the heat is not utilized, the final rejection is made to atmospheric air. In small condensing units, the final transfer may be directly to the air from the dry condenser surfaces, but economy dictates that the principle of evaporative cooling of water in air be employed so that equipment sizes may be kept within practical limits. The water-cooling tower is an apparatus which discards heat from a circulating water stream to atmospheric air by sensible heat transfer and evaporative cooling. Thus, with water acting as the vehicle for heat removal from a condenser or heat exchanger to a cooling tower, economical operation is

attained by recirculating the water, and continuously rejecting the heat to the atmospheric air.

Principle of Cooling by Exposure to Air

2. In **Figure 1,** A, B, and C represent the progressive positions of the temperature of a drop of water during its cooling in air.[1] The inner circle in each case is the drop, while the outer, concentric circle represents the atmosphere of vapor-saturated air which encases the drop. In the atmosphere ring of the drop, dots represent the relative proportions of water vapor present in the saturated air layer existing at the bulk temperature of the drop. The instantaneous rate of evaporation from the water drop is proportional to the weight difference, **per pound of dry air,**[2a] between the water vapor content of saturated air at the water temperature and the actual vapor content of the air. The hotter the water, the greater will be the moisture content in the saturated atmosphere of the water drop, and the greater will be the driving force for evaporative cooling. It is important to note that this driving force depends upon the moisture present **per pound of dry air,** and not upon the dimensions or volume of the atmosphere ring.

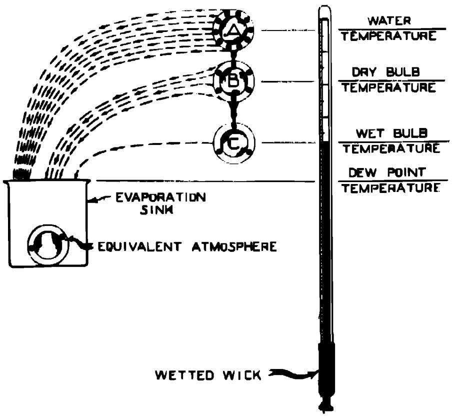

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Fig. 1. Progressive Positions of Water Drop Temperature during Air Cooling

If the air supply for cooling the drop of water were infinite, the moisture content of the air would be unaffected by evaporation from the very small drop, and the condition of an **evaporation** sink would be established. Further, under these conditions, the temperature of the air would be unaffected. The drop of water, at condition A, is cooled both by the flow of sensible heat to the colder air and by the evaporation which occurs when the most active water molecules leave the drop, thus reducing its mean internal energy. If heat is not supplied to compensate for this loss of energy, the temperature of the water drop must decrease. When the drop of water has been cooled to the temperature of the air, position B, sensible heat transfer ceases, because of the absence of a temperature gradient, but evaporative cooling continues if the air is not saturated. At condition A, there are shown 10 dots of relative moisture concentration, and the unbalanced driving force of vapor diffusion is shown by the 8 rays to the evaporation sink. When the drop has been cooled to the air (or dry bulb) temperature, there are but 6 dots of relative moisture concentration in the drop atmosphere, and the moisture diffusion rays to the evaporation sink are reduced to 4. Cooling, however, does not cease since the air is unsaturated. The continuing action of evaporation reduces the temperature of the drop, even though heat from the warmer air flows into the drop, tending to heat it. When the cooling effect of evaporation is offset by the flow of heat from the warmer air, a state of equilibrium is reached, and the temperature of the drop is that of the **wet bulb temperature.**

The equivalent atmosphere of the sink is shown as 2 dots of relative moisture concentration, corresponding to the dew point condition. If the air had been initially saturated at the dry bulb temperature, with its atmosphere of 6 dots, the driving force would have been from 10 dots at A to 6 dots at B, and the relative number of rays would have been 4 for position A instead of the 8 indicated. Furthermore, the dew point and the dry bulb temperatures would have been the same, and the wet bulb temperature would necessarily have been equal to the dry bulb temperature. It is most important to note, however, that cooling would still have proceeded, since both a temperature differential and a vapor concentration gradient would have existed at the start of the cooling.

In actual cooling towers, the air supply is not infinite. This air supply lacks the capacity of a sink to completely engulf the evaporation feed, and the moisture content of the equivalent atmosphere increases as the air passes through the apparatus. Therefore, less temperature differential from air to water is required to arrest the temperature lowering due to evaporation,[1] and the wet bulb temperature rises as the air passes through the cooling tower.

The wet bulb temperature[3] may be accepted as the lowest temperature to which water may be cooled in air. Furthermore, analysis has demonstrated that the equations of water cooling or heat surrender may be based upon driving forces calculated between properties of the water temperature and the wet bulb temperature; the consideration of the dew point and the dry bulb temperatures need not enter into the determination of the driving forces. In fact, a very convenient method of analysis utilizes a driving force which is the difference between a heat content of saturated air at the water temperature minus a heat content of saturated air at the wet bulb temperature[4] of the air. (The heat content quantity used may be enthalpy or sigma function.) Whenever heat is added to air, the wet bulb temperature increases; this is true even though no water vapor is added. Consequently, when the heat from a refrigeration system is rejected into air by a cooling tower, the wet bulb temperature of the air passing through the cooling tower is increased and the driving force for cooling is decreased.

A study of Figure 1 indicates that evaporative and sensible heat transfer occur simultaneously. Fortunately, the conductance[5] (over-all heat transfer coefficient) for sensible heat transfer through the air film to the water surface is directly proportional to the coefficient of vapor transfer for evaporation, and the heat and evaporation equations may be combined, using the conductance for sensible heat transfer through a dry air film as a basis. Thus, arrangements of surface which give the best conductances for tube bundles in dry air are indicated as desirable guides in the arrangement of the water-covered guiding surfaces of cooling towers. For instance, with respect to conductance, staggered tubes[6a] in banks are superior to tubes in line, tubes in banks are superior to single tubes, counter flow is generally superior to cross flow,[6b] and cross flow is generally superior to parallel flow.

Balancing Factors

3. **The driving force** is a function of the difference between the temperature of the water and the wet bulb temperature of the air in contact with the water. As air quantity is decreased in a given cooling tower for a given heat removal and a constant wet bulb temperature of entering air, the final wet bulb temperature of the air is increased, since the wet bulb temperature increases as heat is added to the air. Under such conditions, the driving force for cooling is decreased.

4. **The surfaces for heat transfer** in a water cooling tower comprise the moving water films flowing over the guiding filling surfaces,[7] the drops, the streams of water, and all other surfaces which are covered with water, including water surface in the basin and water films on structural members. The **basic loading,** $\overline{\text{GPM}}_b$, is equal to the total flow of water over the cooling tower in gallons per minute divided by the active horizontal area of the tower. The active horizontal area is the horizontal cross sectional area of the cooling tower in square feet across which the water passes in its gravity flow from the top to the bottom of the cooling tower. In an atmospheric tower, the active area is the internal width, face to face of posts multiplied by the tower length, center to center of end posts; and it is always area inside the louvers. In a mechanical draft tower with counter flow of air and water, the active area is the gross area of the airway, and corresponds to the face velocity directed against a coil. For a constant construction pattern, a decrease in $\overline{\text{GPM}}_b$ allows a greater prorata of surface per gpm, but the coverage of water over the surfaces is decreased at light loadings due to channeling.

5. **The heat transfer coefficient** for cool-

ing tower action varies with the relative velocity of air over the film surfaces. There is strong evidence[6c] to show the presence of a relatively slow-moving, streamline film of air on the surface of a solid over which a stream of air is flowing. The heat transfer through such a film is by conduction until the heat penetrates the film. The heated molecules at the boundary of the film are then picked up and carried away by the swirling action of the main body of the fluid, and the heat transfer is then mainly by convection. As the velocity of the air stream is increased, the thickness of the streamline layer is decreased by the increased wiping and rubbing action of the air stream, and the insulating resistance of the film is reduced, allowing a greater heat transfer or conductance through the film. Thus, as in the case of any insulation problem, the reduced thickness of insulation permits greater heat flow, and increased air velocity over the cooling surfaces is associated with a greater rate of heat exchange.

6. The cooling tower capacity is indicated by the rate of water circulation and the water temperature, and is controlled[7] by:

a. Driving force dependent upon wet-bulb temperature and air quantity.
b. Heat transfer coefficient as determined by the air velocity and character of the surface.
c. Amount of cooling tower surface.

It is very important to understand that, irrespective of the amount of surface in the cooling tower, the air quantity may be so low in relationship to the rate of water circulation that a theoretical point of zero driving force[8a] may develop within the tower path. Under such a condition, water temperatures must increase in order to allow positive driving forces to exist throughout the tower. When the ratio of air to water is too low, addition of surface will not correct the difficulty. (See Figure 15.)

Design Conditions

7. The wet bulb temperature of design should not be exceeded more than 5 per cent of the operating hours during a normal summer. Atmospheric (free convection or natural draft) equipment should be designed to operate properly with no more than one-half of the average wind velocity, and the maximum design wind velocity should not exceed 5 mph.

8. Evaporation will be about one per cent of the water circulated for each 10 deg F of water cooling range, provided the cooling is mostly evaporative. Since the condenser heat output to the cooling water is approximately 15,000 Btu per hour per ton of refrigeration, and since the heat to vaporize one gallon of water is about 8330 Btu per gallon, a cooling tower must evaporate, on the average, 15,000/8330, or 1.8 gallons **per hour** per ton of refrigeration to the compressors. **Figure 2** may be used for estimating the condenser heat rejection to the cooling tower water. In using Figure 2, it is important that the saturated suction gas temperature be that of the compressors connected to the condenser. Also, the refrigerating effect must be applied to the capacity of the compressors discharging directly to the condensers and not to the capacity of the low stage or booster compressors. This is necessarily so, since the high pressure compressors must handle not only the pay load from refrigerated space or processes, but also the heats of compression of the low temperature ma-

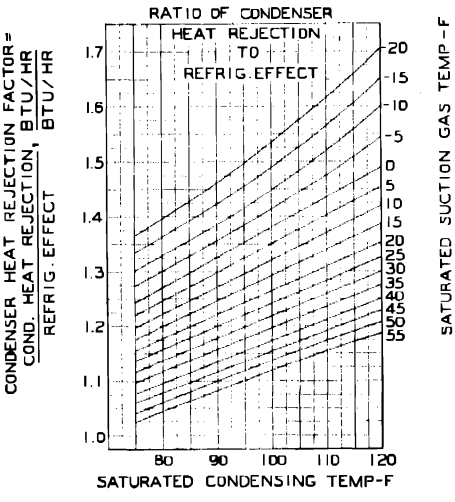

Fig. 2. Chart for Estimating Heat Rejection from Refrigeration Condensers (Mechanical Refrigeration)

chines, the refrigerating loads of sub-cooling, etc. In such compound systems, the proper suction pressure is intermediate between the suction temperature of the low pressure machine and the discharge pressure of the high pressure machines.

9. **Continuous blow down** (or bleed) is generally necessary in cooling tower circuits. Since makeup water must be continually supplied to offset the approximate 1.8 gallons per hour per ton of refrigeration, this makeup water will carry into the system the mineral concentration of its analysis. When water has been vaporized in the process of evaporative cooling, it has actually been distilled from its original salts, and the concentrations increase in the cooling tower circuit. A point may be reached at which heavy deposition of scale will occur. Therefore, means must be provided for blowdown from the circuits of cooling towers and spray ponds. For instance, if 1000 ppm of salts were the maximum which might be held in a system for good operation, a makeup water of 1000 ppm would not be allowable. A simple mass balance will give the relationship between blowdown water and evaporated water as follows: The ratio of **blowdown** water to **evaporated** water is equal to the ppm of the makeup water divided by the difference between the ppm for the system and the ppm for the makeup water. For example, with a makeup water having 250 ppm salts and a system water satisfactory at 1000 ppm salts, the ratio of the blowdown water to the evaporated water is equal to $250/(1000-250)=\frac{1}{3}$; therefore, $\frac{1}{3}\times 1.8=0.6$ gallons **per hour** per ton of refrigeration blowdown. The makeup is then equal to the evaporation plus the blowdown; or in the case of the example: $1.8+0.6=2.4$ gallons per hour per ton of refrigeration.

Atmospheric cooling towers are subject to the actions of cross winds, and in many cases will achieve blowdown without actual drainage from the system. In no case should the makeup water allowance to cover windage be less than one per cent; and in high winds, the amount may run considerably greater. Mechanical draft cooling towers are provided with very efficient drift eliminators which limit the loss by entrainment in the air stream to figures approximating 0.3 per cent of the circulated water. Towers of this type must be definitely provided with blowdown in order that excessive scaling may not occur within the tower or on the surfaces of the heat exchange equipment used with the cooling tower water.

Structure

10. The cooling tower in a large plant is a structure of considerable proportions. The shipping weights of wooden cooling towers are rarely the operating weights, since the wood picks up moisture during operation. Many cooling towers are constructed of materials which will develop weights under operation as high as 5 pounds per board foot of lumber when saturated. Therefore, in designing supporting structures, weights of saturated parts should always be considered.

The tower generally exposes a considerable surface to wind pressure. Also, there are many cases in the western states[9] where seismic lateral loads must be considered. Where lateral and wind loads develop, the pressures on the leeward posts are increased due to the overturning action of the lateral forces, and such loads must be properly applied to the dead weights of the tower in designing foundation structures.

11. **Water treatment** is generally necessary for satisfactory cooling tower operation. The build-up of scales and the fouling[7] due to algae and slimes should be investigated for each water and locality; and proper means of control should be adopted. Refrigerating Engineering Application Data—Section 46 of the American Society of Refrigerating Engineers is a valuable guide in this matter.

12. **Approach to the wet bulb** is an important design quantity. It is the difference between the temperature of water leaving the cooling tower and the wet bulb temperature of air entering the cooling tower. Since the wet bulb temperature is the lowest temperature to which water may be cooled in air, a theoretically infinite size of cooling tower is required to cool water to the wet bulb temperature. Thus, as the approach to the wet bulb is decreased for a given cooling load, the cost of the cooling tower installation is increased. It is important that the combina-

tion of the condenser and the cooling tower be economical. In many cases, an approach of 7 to 10 F will be satisfactory. A very excellent analysis on this subject is given by Degler.[10] The combination of condenser and cooling tower should be considered for economical operation between the limits of the wet bulb temperature and the condensing temperature. Arbitrary selection of a low approach to the wet bulb may be quite unsatisfactory from an economic standpoint.

Atmospheric Cooling Towers

13. The atmospheric splash-deck cooling tower is illustrated in typical construction in **Figure 3.** Towers of this type are known as free convection towers, since the air currents are not supplied by mechanical means. The air supply may be principally due to the action of wind currents, or it may be due to the thermal air currents developed by temperature and density differences generated during cooling action. Water is distributed at the top of the tower, and passes downward to the basin. The distribution system shown in Figure 3 is a low pressure, splash-impact system. However, other systems that are used are the trough and the spray distribution types. In some cases, sprays are directed upward. Where such spray distributions are enclosed within solid fences, the efficiency is greatly decreased due to the lack of air supply to the spray chamber. A superior result is obtained by having the sprays directed downward into adequately louvered space, thus allowing the removal of heat by cross travel of the air.

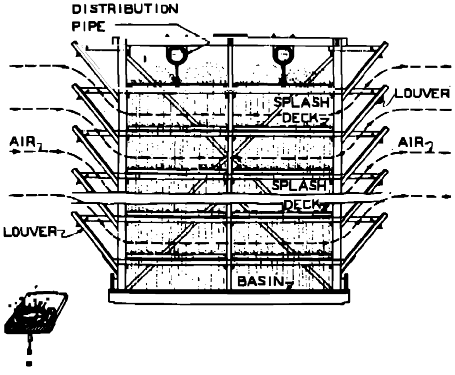

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Fig. 3. Atmospheric Cooling Tower

During the gravity flow of the water from the top of the tower to the basin, the water covers the guiding surfaces of the slats as shown in Figure 3. Furthermore, all of the structural members offer guiding surfaces for the establishing of water-film transfer area. The falling and splashing of the water offers some surface and stirring action. With a given wind supplying air to the louver system of a tower of given height with broadside exposure, a short, wide tower will intercept less air than will a long, narrow tower. Therefore, for a given heat load, tower height, and water flow, the average wet bulb of the shorter tower will be higher than that of the longer tower; and the cooling range of the water on the shorter tower must be at a higher position on the thermometric scale than is the case for the longer tower. The economic balance between available natural air currents and structure size for an atmospheric tower dictates, to some extent, the arrangement of the surface. However, the capacity of an existing tower may often be increased by the addition of intermediate decks, properly arranged and proportioned;[7] and by the addition of efficient spray chambers at the top.

Sagging deck surfaces will not develop full water-film surface, and the water streams will be concentrated in a number of fractions of tower length, with a part of the air by-passing the active areas. The resulting effect is that produced by a number of small towers[7] with relatively great width and reduced water-film surface per gallon cooled. Since decreased area reduces heat transfer and reduced air flow reduces driving force, the effect of sagging guiding surfaces of the decks is to cause the position of the cooling range to rise in the thermometric scale for a given heat load, water flow, wind, and wet bulb temperature, thus giving higher water temperatures, higher condensing temperatures, and less economical plant operation.

14. Spray or mist type water cooling towers operate principally with drop-surface exposure. Thus, if no splash decks are provided within a tower louvering system, cooling may be accomplished by the action of spray nozzles spraying downward from

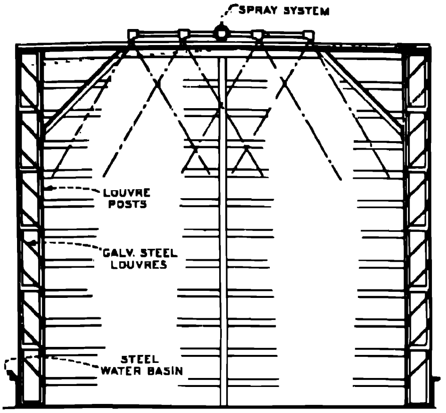

Fig. 4. Atmospheric Spray Tower Equipped with Louvres to Reduce Water Loss Due to Windage

the top of the tower. **Figure 4** shows an atmospheric spray tower. The operation of spray towers is controlled by the characteristics of the nozzles. When water is sprayed into the air, some of the very fine drops are completely vaporized,[11] and therefore contribute nothing to the cooling of the circulated liquid from which they are sprayed. The liquid within the drops is not subjected to mechanical stirring. Heat must pass from the heart of the drop to the film by means of convection and conduction. The greatest amount of cooling occurs where the water enters at spraying, due, somewhat, to the high conductance or heat transfer coefficient which accompanies the high velocity of projection from the nozzles. The walls and framework of the spray chamber add to the film surface exposed to the air movement.

In spray towers, some air supply is generated by the aspirating action of the nozzles. Therefore, spray towers are less sensitive to variation in wind velocity than are splash-deck atmospheric towers. However, spray towers do not generally have the capacities of spash-deck towers.

Long temperature ranges and close approaches to the wet bulb generally require splash-deck towers because of the greater areas that may be developed per unit of tower volume. Spray towers are quite satisfactory, in many cases, for air conditioning applications.

15. Screen-deck mist towers[12] may be used where a high rate of windage loss is not objectionable from either the standpoint of economy or the standpoint of drift nuisance. This type has been used on refrigeration plants with refrigerating capacities from 15 to 1000 tons. The tower is built with louvered sides and ends as in the construction of the atmospheric splash-deck tower. The decks may be ½ in. mesh, 18 gage galvanized steel wire cloth stretched over 2×4 in. supports not over 4 ft apart. Water may be distributed from a perforated flume or from a pipe system with low pressure nozzles through which the water bubbles. In either case, the streams fall onto a splash board before contacting the screens. The screens are placed at 18 in. centers vertically. Towers with more than 15 decks experience heavy drift loss at the bottom with medium winds. When the broadside wind exceeds 10 mph, the drift loss is excessive. Since the water of a screen-deck tower does not experience the high velocity of distribution of a spray tower, the effect of the wind velocity upon the sensitivity of the tower is more marked than in the case of a spray tower.

16. Spray ponds cover a greater area per gallon per minute circulated than do spray towers. **Figure 5** shows a layout for a basic single spray header, with the spacing indicated for headers added to leeward. A louvered fence with its top at a height not less than the spray height should be used when the distance to the edge of the pond area is less than 25 ft. The nozzles may be installed in staggered

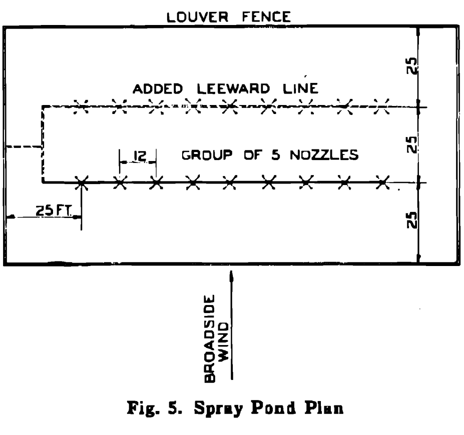

Fig. 5. Spray Pond Plan

arrangements as well as the clustered arrangements shown in Figure 5. Nozzles generally operate quite satisfactorily at a pressure of 7 psi. The height of the spray above the nozzles will then be about 7 ft, and the height of the spray above the water surface will be about 12 ft.

Sizing of Atmospheric Cooling Towers and Spray Ponds

17. Figure 6, the Basic Sizing Chart for Water Cooling Towers, gives relationship between tower size and water temperature for various wet bulb temperatures of air entering an atmospheric cooling tower. The sizing scale was originally computed for a basic atmospheric cooling tower having decks spaced vertically at 2.5 ft; a basic loading, $\overline{\text{GPM}_b}$, of 1.5 gpm per sq ft; a broadside wind velocity of 5 mph against the tower with louvers inclined to the horizontal at an angle of 52 deg; and an active width of 12 ft. The relationships between the temperature scale and the sizing scale were developed by a number of arithmetic step calculations to give the integrated curves. The relationships for other deck spacings, tower widths, basic loadings, wind velocities, etc., were then developed by field tests and bracketing correlation. To utilize the chart, determine the sizing scale values S for the temperatures on tower and off tower respectively at the wet bulb temperature of air entering the tower. Compute the difference between the sizing scale values for the two water temperatures in question. The difference, ΔS, is then proportional to the size of the atmospheric cooling tower. The sizing scale intercept ΔS is then used with the following tables and charts to determine approximate tower sizes.

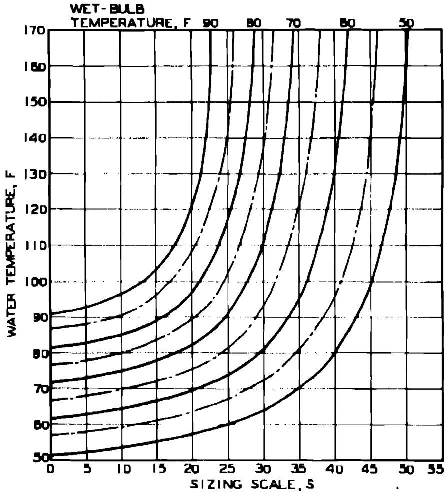

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Fig. 6. Basic Sizing Chart for Water Cooling Towers

18. Atmospheric splash-deck towers exposing approximately 2.7 sq ft of wetted surface area per deck per sq ft of active horizontal area of tower, with average louver constructions for a louver inclination of 55 deg from the horizontal may be estimated by calculating the number of splash decks, using the equation:[7]

$$N = r(\Delta S)\,\frac{W}{m}\,\sqrt{\frac{30}{B}} \tag{1}$$

The factors, r and m, are determined from the alignment charts of Figures 7 and 8. Thus, for a tower width of 10 ft, $\overline{\text{GPM}_b}$ of 1.8 gpm per sq ft, and a deck spacing of 12 in. center to center vertically, r is 4.3 and m is 2.2.

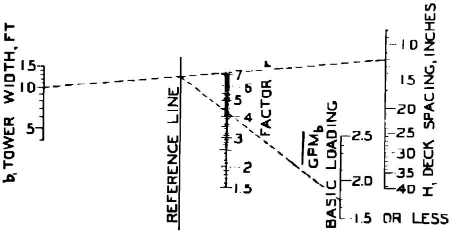

Fig. 7. Alignment Chart for Evaluating *r* in Equation 1

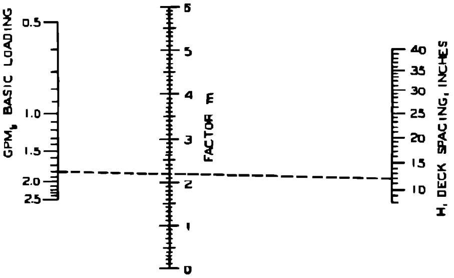

Fig. 8. Alignment Chart for Evaluating *m* in Equation 1

Table 1. Values of Wind Velocity Factor

Broadside wind velocity, mph	Factor W
0	2.10
1	1.70
2	1.44
3	1.26
4	1.11
5	1.00
6	0.93

In cases where N is known and ΔS is the unknown, the equation may be solved for ΔS. The value of ΔS, the sizing scale intercept, may then be applied against the wet bulb curve in question to determine the possible terminal temperatures for the cooling tower under consideration.

The coefficients of heat exchange for free convection for surfaces into air are functions of a large number of variables. The transfer coefficients for natural convection[7] involve the mean temperature difference, dimensional characteristics of shape, orientation of the heated surface, etc. In general, the factors given for W in **Table 1** will be satisfactory for refrigeration service.

Table 2. Volumes for Atmospheric Spray Towers

b Active width ft	$\overline{GPM}_b$ Basic loading gpm/sq ft	Volume of spray space cu ft/gpm
6	1.95	5.80
8	1.70	8.00
10	1.52	9.60
12	1.38	11.0
14	1.25	12.2
16	1.14	13.1

Table 3. Values of Sizing Scale Intercept for Atmospheric Spray Towers

t_1 Final temperature F	ΔS Sizing scale intercept decks
50	8.2
60	7.6
70	7.1
80	6.7
90	6.4

19. Atmospheric spray towers[13] (with broadside winds of 3 to 5 mph, and with efficient nozzles operating at 7 psi) sized from **Table 2** will have the values of ΔS indicated by **Table 3** (30 inch barometer). The value of the sizing scale intercept, ΔS, may be used with the basic sizing chart of Figure 6 to determine the possible water temperature ranges at various wet bulbs and terminal temperatures.

20. Screen-deck mist towers[13] should have an active tower volume of 15 cubic feet per gpm circulated with an active width of 12 ft. An average loading of $\overline{GPM}_b$ is 1.25 gpm per sq ft; and the loading should not exceed 1.50. When a tower is designed with these size specifications, the value of ΔS for a 30 inch barometer is given in **Table 4**.

Table 4. Values of Sizing Scale Intercept for Screen-Decked Mist Towers

t_1 Final temperature F	ΔS Sizing scale intercept decks
50	9.0
60	8.3
70	7.9
80	7.5
90	7.1

21. Spray pond performance[14a] may be estimated by the use of **Table 5**. The data are for a single line of nozzles with the nozzles set in clusters of five, a broadside wind of 5 mph against the louver spray fence, a barometric pressure of 30 inches Hg, and flow rates of 25 and 40 gpm per nozzle at 7 psi. A typical spray pond layout is shown in Figure 5.

Table 5. Values of Sizing Scale Intercept for Spray Ponds with a Single Line of Nozzles

Gpm per nozzle at 7 psi	Final temperature, t_1, F				
	50	*60*	*70*	*80*	*90*
25	10.2	9.5	9.0	8.5	8.1
40	8.3	7.4	6.8	6.3	5.9

If a leeward line of nozzles is added as indicated by the dashed lines in Figure 4, the performance may be estimated as follows:

a. With a height of spray above the water surface of 12 ft, each group of nozzles has an air supply through a vertical area of 144 sq ft. The air velocity across the louvered pond may be taken at 330 fpm for a broadside wind of 5 mph; and the air supply per nozzle cluster is therefore 47,500 cfm. The weight of the air per cluster of nozzles is found by dividing this amount by the humid volume of the air in cubic feet per pound.

b. With a given temperature t_2 of water to the spray nozzles, evaluate the temperature off the first line of nozzles from Table 5, using the wet bulb temperature of the air entering the pond through the first row of nozzles, and an approximation of t_1 for the selection of ΔS.

c. Divide the weight of water sprayed per nozzle cluster by the weight of air flowing per nozzle cluster, and multiply the result by the change in temperature of the water as evaluated in (b). The result is the increase in the heat content of the air per pound of dry air, and may be evaluated either as enthalpy or sigma function for practical purposes (see psychrometric tables, Chapter 10).

d. Add the heat content increase of (c) to the sigma function or enthalpy of saturated air at the temperature of the initial wet bulb to find the sigma function or enthalpy of the wet bulb temperature of the air leaving the first row of spray nozzles. Identify this leaving wet bulb temperature from the psychrometric tables by finding the temperature of saturated air which corresponds to the sigma function or enthalpy so calculated.

e. With the wet bulb temperature determined in (d), and using the data of Table 5, determine the leaving water temperature of the second row of nozzles, employing the common temperature of t_2 to enter the basic sizing chart.

f. Since the nozzle lines are identical and fed from a common supply, the resultant final temperature of the spray pond water will be the average of the final temperatures of the two lines; and the cooling range will be this average temperature subtracted from the common temperature t_2 which feeds the nozzles.

Forced Convection Cooling Towers

22. Atmospheric cooling towers are principally dependent upon wind currents for operation. In many installations, the forced convection cooling tower is used, since it is not dependent upon wind currents for air supply and is therefore independent of the wind variable. Forced convection cooling towers operate with mechanically forced or induced draft. When the air is forced through the tower from fans at the bottom of the tower, the tower is said to be of the mechanical draft type. When fans are mounted at the top of the tower and exert a suction on the tower in drawing the air through, the tower is said to be of the induced draft type. **Figure 9** shows an induced draft cooling tower in which the guiding surfaces for water-film generation are provided by a relatively dense filling. In the case of Figure 9, air is drawn into the tower at the base through the inlet. The inlet is often constructed of downward inclined louvers; in the case of Figure 9, an airfoil type inlet is shown which reduces the air shock losses in pass-

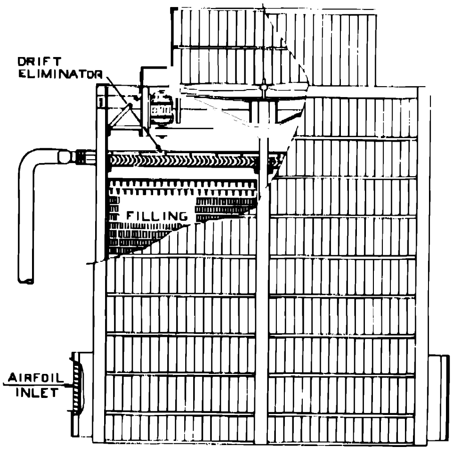

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Fig. 9. Induced Draft Tower

ing through the inlet. After the air has entered horizontally through the inlets, it passes upward vertically over the film surfaces generated by the filling. Before passing to the fan chamber, the air must pass through drift eliminators which arrest entrained water.

Where low pressure distribution is used from troughs or pipes operating under very slight head, 2 pass eliminators are satisfactory. If spray distribution is used, 3 passes are generally advisable. The fan is typically a propeller type with airfoil blades. In small sizes, the fan may be on a direct drive mount on the motor. In intermediate sizes, the V-belt arrangement of Figure 9 is very satisfactory. On large sizes, running up to 16 ft fan diameter, the fan is generally mounted on a gear reducer which is operated by a right angle drive to a standard motor outside the fan fence. Propeller fans are used in water cooling towers because they move high air quantities at low static pressures with high fan efficiencies.

23. **Distribution** is effected by means of troughs, flooded deck areas, splash-impact systems, and spray systems. Trough systems allow the gravity overflow from troughs for distribution or the flow of water from troughs through holes and notches. Troughs must be placed to allow splash which will give even and adequate coverage to the distributing deck areas. Flooded deck distribution systems are really large, shallow tanks, the floors of which are perforated with holes or nozzle outlets. Water is allowed to flood the complete active tower area, and passes through the bottom outlets in uniform distribution. Splash-impact systems function by the discharge of water downward from header pipes operating at slight pressures which are great enough to require a complete pipe housing to maintain and generate the discharge pressures; and the impact of the water streams on appropriate splash strips gives the spread to the water. Spray systems are nozzle systems which may spray downward or upward in the spray chamber. When a spray system is used without filling, the tower is known as a spray-filled tower, or merely as a spray tower. The pressure necessary to operate sprays must be added to the static lift of the distribution header to arrive at the total static lift for pumping computations.

24. **Air quantities** in the forced convection cooling towers are very much lower per pound of water circulated than they are in atmospheric or free convection cooling towers. A wind velocity of approach of 5 mph against one foot of length of an atmospheric tower 12 ft wide and 30 ft high will pass through ordinary louvering at approximately 340 fpm; an air supply of 10,200 cfm through the one foot of tower length. Ordinarily, the water loading of an atmospheric tower is held at from 1.0 to 2.0 $\overline{\mathrm{GPM}}_b$, with a good average value at 1.5. At that figure, the flow would be 18 gpm, discarding heat to 10,200 cfm; or 150 pounds of water discarding heat to approximately 750 pounds of air, a ratio of water to air of 0.20. In forced convection towers, however, the water-air ratios run from 0.75 to 1.3 pounds of water per pound of dry air. The forced convection cooling tower, therefore, operates with decreased driving forces due to the increased average wet-bulb temperatures which develop due to the economically restricted flow of mechanically supplied air. The deficiencies in driving forces, in comparison with atmospheric towers, must be overcome by increased area and increased heat transfer coefficients. Compact, induced draft cooling towers, such as indicated in Figure 9, provide the necessary heat transfer requirements, satisfactorily carrying water loads of from 2.0 to 6.0 $\overline{\mathrm{GPM}}_b$. In general, a good balance in tower operation occurs when the gross or face velocity of the air against the packing is 100 times the value of $\overline{\mathrm{GPM}}_b$. Thus, with a basic loading of 4.0 $\overline{\mathrm{GPM}}_b$, a generally balanced design will occur with a velocity on the gross cross sectional area of the tower A of 400 fpm. As velocities increase, the pressure losses through the packing due to the frictional turbulances increase about as the ratio of the squares of the velocities involved.

Design of Forced Convection Cooling Towers

25. **The cooling equation**[4,14b] for the counter-flow, forced convection cooling tower may be written as follows:

$$8.33\ \overline{\mathrm{GPM}}_b(t_2 - t_1) \cong f_d a_s H_f (\Sigma_t - \Sigma)_M \quad (2)$$

The conductance f_d may be evaluated by the study of the filling arrangement and the formula method given in this chapter. The value of a_e, the equivalent active area per cubic foot of tower, may be evaluated by actual computation of the exposed filling area corrected by a coverage factor, plus the dynamic area effect of water falling and splashing, given in this chapter. The chart of Figure 6 may be used in conjunction with the charts which follow to determine values of $f_d a_e H_f$ of Equation (2). With the values of f_d and a_e determined, it is then possible to solve for H_f, the height of filling in the tower construction in question, at various values of gross air velocity and basic water loading.

26. The Wet Bulb Factor Curves[8b] of **Figure 10** are used with the chart of Figure 6 for calculation of mechanical draft, forced convection cooling towers. As explained, the basic sizing chart was originally designed for atmospheric cooling towers in which an increase in the sizing scale intercept ΔS is accompanied by an increase in tower height and therefore an increase in air quantity. In the mechanical draft cooling tower, air quantity due to wind does not increase as the tower height is increased. The sizing scale intercept ΔS is used to enter Figure 10 at the appropriate wet bulb temperature of air entering the cooling tower to determine the wet bulb factor F. This factor is then multiplied by ΔS to obtain the sizing index I.

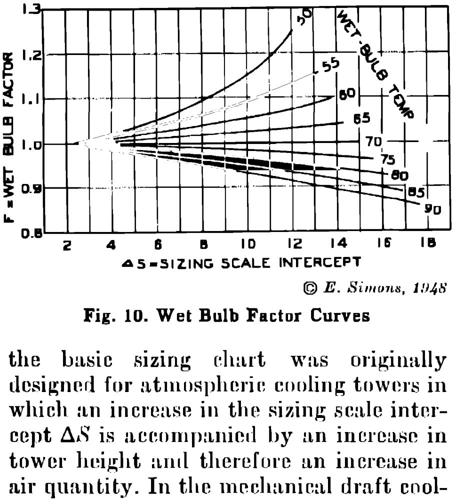

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Fig. 10. Wet Bulb Factor Curves

27. The values[15] of $f_d a_e H_f$ at 30 in. Hg barometer are obtained from **Figures 11 through 15** inclusive, which cover a range of GPM_b from 2 to 6 gpm per sq ft of basic area at various air rates. It is then possible to compute H_f when f_d and a_e are known; or, oppositely, it is possible to arrive at the value of $f_d a_e$ for a given filling when the height of the filling is known and the air rate, basic loading, water temperatures, and wet-bulb temperature are known. Also, by the use of Figures 6, 10,

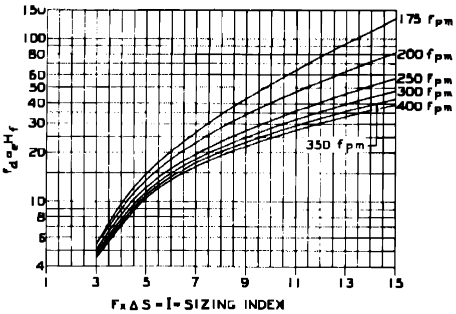

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Fig. 11. $f_d a_e H_f$ versus Sizing Index at Basic Loading of 2 gpm per sq ft

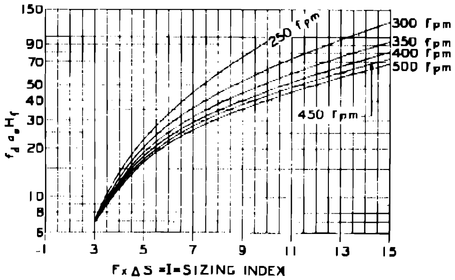

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Fig. 12. $f_d a_e H_f$ versus Sizing Index at Basic Loading of 3 gpm per sq ft

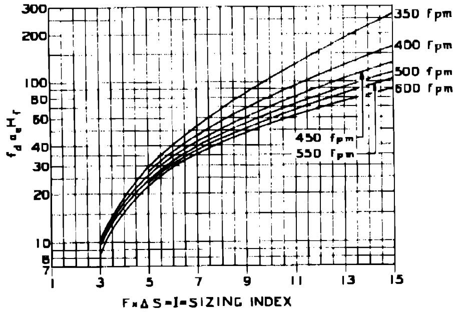

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Fig. 13. $f_d a_e H_f$ versus Sizing Index at Basic Loading at 4 gpm per sq ft

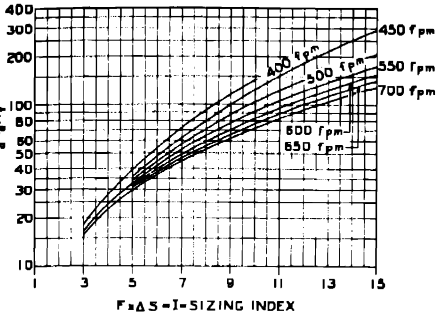

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Fig. 14. $f_d a_e H_f$ **versus Sizing Index at Basic Loading of 5 gpm per sq ft**

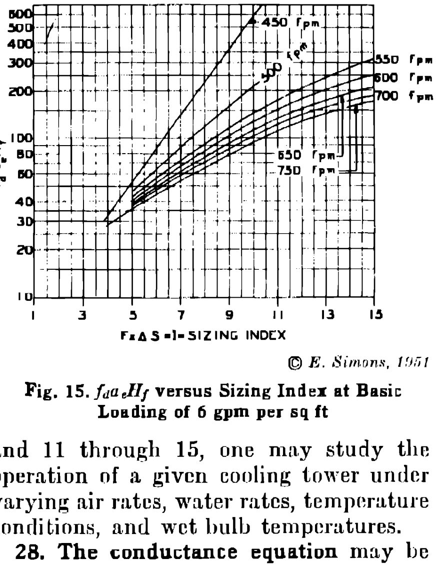

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Fig. 15. $f_d a_e H_f$ **versus Sizing Index at Basic Loading of 6 gpm per sq ft**

and 11 through 15, one may study the operation of a given cooling tower under varying air rates, water rates, temperature conditions, and wet bulb temperatures.

28. The conductance equation may be estimated from the Lewis modulus, the filling arrangement, and data from McAdams.

The Lewis equation holds reasonably well for the evaporation of air into water. This equation[5,14c] may be stated in the following form:

$$f_c \cong f_d s \tag{3}$$

McAdams[6d] cites Winding's[16] findings on staggered banks of steel tubes. A bank of staggered steel tubes 8 rows deep gives substantially the same results whether the tubes are circular, oval, or streamline in shape. The tubes in the Winding experiments had a perimeter of 4.7 in. McAdams therefore concludes that the data may be correlated in terms of the heated length[6e] exposed to the air stream. Further study of the data indicates that, in the case of a single row of tubes, the average heat transfer coefficient is about 67 percent of the coefficient which exists in any row of a tube bank from the third row on. The second row has a heat transfer to air of about 83 percent of that of the third row. Therefore, when the guiding surfaces for the filling of a cooling tower are placed in a staggered arrangement similar to that of a staggered tube bank, the highest rate of heat transfer may be expected. When the perimeter of a filling slat is divided by π, the result is the diameter of an equivalent circular pipe. The dry heat transfer coefficient for that pipe may then be computed by certain well established formulae. This procedure for the calculation of the dry conductance has been satisfactorily checked in actual installations; it is based upon the experimentally established fact that heat transfer coefficients may be correlated on the basis of the heat transfer length or perimeter of a pipe or fin in an air stream. McAdams equation[6f] may be combined with the Lewis modulus to obtain a conductance formula in terms of the perimeter of the filling slat placed across the air stream. Since the water streams in a cooling tower move downward under gravity action against the upward air movement, the heat transfer coefficient will be higher than for a stationary surface. For practical cases, the value computed from the McAdams formula may be increased by 14 percent to allow for the effect of the moving water surfaces. The result so obtained will approach the average upper limit of the values for staggered tubes given by McAdams.[6g] With the perimeter of the filling element in inches, the conductance equation is as follows:

$$60 f_d s \cong \frac{1.14 \times 0.133 s \left(\frac{60 v_{\max}}{V_a}\right)^{0.6}}{\left(\frac{P}{12\pi}\right)^{0.4}} \tag{4}$$

Equation (4) reduces to the simplified form of

$$f_d = \frac{0.126}{P^{0.4}} \left(\frac{v_{\max}}{V_a}\right)^{0.6} \tag{5}$$

29. Conductance in spray-filled forced convection cooling towers is locked to the equivalent area per unit volume of tower in the expression $f_d a_e$. With centrifugal nozzles discharging downward at the rate of 3 gpm per nozzle under a pressure[14d] of 7 psi at the nozzles, an overall value (including spray and walls) of $f_d a_e$ may be obtained from **Table 6.** Poor, inefficient nozzle action may reduce the values of $f_d a_e$ to 80 percent of the values of Table 6.

Table 6. Values of $f_d a_e$ for Forced Convection Spray-Filled Towers ($t_i = 80$ F)

H_f ft	$f_d a_e$
4	2.07
6	1.78
8	1.61
10	1.47
12	1.37
14	1.30
16	1.24

30. The effective area per unit volume a_e comprises the actual film area on the filling surfaces plus area developed by the dynamic action of the water falling and splashing. The area of the filling exposed to the air flow a_f may be computed from the filling pattern of the cooling tower. When the area of the filling is too great for the water flow, the water will tend to channel, and the film surface will break, allowing unwetted filling area which is not active in cooling. Furthermore, with a given water flow over a given horizontal filling arrangement, channeling and loss of wetting will occur as the height of the individual filling pieces is increased. The cycle modulus[4] C_z has offered a satisfactory parameter for the identification of the filling coverage.

If, in a symmetrically filled[14e] tower, the water passes through a first horizontal section at right angles to the direction of flow and continues through the tower until it passes a second horizontal and parallel section which cuts the filling so that the sectional elements of the cut filling are in perfect alignment and typical location with respect to similar elements cut by the first section, the water has passed through the height of one cycle. When the height of the cycle is z, in inches, the equation of the cycle modulus (an empirical parameter) is

$$C_z = \frac{12\overline{\mathrm{GPM}}_b}{a_f z} \qquad (6)$$

With the value of C_z determined from Equation 6, the percentage of wetted filling area α may be determined from **Table 7.** The dynamic splash effect per cycle may be determined from **Table 8.**

Table 7. Percent Wetted Filling Area vs. Cycle Modulus

C_z Cycle Modulus	α, % Wetted Filling Area
0.1	42
0.15	60
0.20	77
0.25	82
0.30	97
0.35	99
0.38 & up	100

Table 8. Dynamic Splash Effect per Cycle

e Splash Height in.	d Splash Effect sq ft/gpm
5	0.35
6	0.40
7	0.45
8	0.50
9	0.55
10	0.59
11	0.64
12	0.68
15	0.83
18	0.94

All quantities are then available to determine a_e, the effective active tower area per unit volume, through the solution of the following equation:

$$a_e = \frac{\alpha a_f}{100} + \frac{12d \times \overline{\mathrm{GPM}}_b}{z} \qquad (7)$$

Examples of Forced Convection Tower Design

31. Figure 16 is a vertical cross section of one of the numerous arrangements possible for SIPAC[8c,17] filling. The water flow is downward over the filling while the air flow is upward against the water flow.

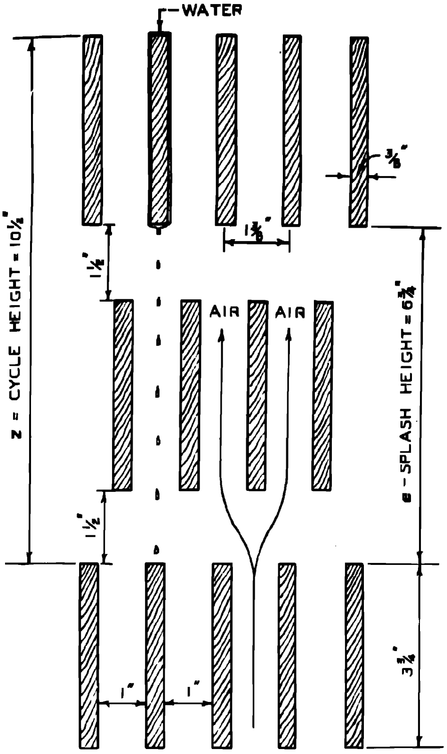

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Fig. 16. SIPAC Filling

Example 1. With the filling arrangement of Figure 16, what is the height of filling required under the following conditions: (1) temperature of water onto tower = 90 F; (2) temperature of water off tower = 80 F; (3) wet bulb of air entering tower = 75 F; (4) basic loading of water, $\overline{\text{GPM}_b} = 3.0$ gpm per sq ft; (5) vertical face velocity against filling $v = 400$ fpm; (6) barometric pressure = 30 in. Hg; (7) $V_a = 14.1$ cu ft per lb dry air?

Solution: Examination of Figure 16 indicates that the perimeter of the filling slat is 8.25 in. The center to center spacing of slats is 1.375 in., while the distance from the top of one layer of slats to the top of the next layer is 5.25 in., one half the height of a cycle. Therefore,

$$a_f = \frac{8.25}{12} \times \frac{12}{1.375} \times \frac{12}{5.25} = 13.73 \text{ sq ft per cu ft}$$

From Equation 6,

$$C_s = \frac{12 \times 3.0}{13.73 \times 10.5} = 0.25$$

From Table 7, $\alpha = 82$ percent. For a splash height e of 6.75 in., Table 8 shows an interpolated value of $d = 0.44$. Therefore, a_e is computed from Equation 7 as follows:

$$a_e = \frac{82 \times 13.7}{100} + \frac{12 \times 0.44 \times 3.0}{10.5}$$

$$= 12.7 \text{ sq ft per cu ft}$$

From Equation 5,

$$f_d = \frac{0.126}{8.25^{0.4}} \left(\frac{400 \times 1.375}{14.1 \times 1.00}\right)^{0.8}$$

$$= \frac{0.126 \times 9.00}{2.33} = 0.487$$

(Note: The ratio 1.375/1.00 is the contraction factor based upon the filling clearances at right angles to the air flow. This ratio, multiplied by the face velocity of 400 fpm, is $v_{\max}$.)

From the chart of Figure 6, using the wet bulb curve for 75 F, $S_{90} = 20.4$ and $S_{80} = 10.4$. $\Delta S = 20.4 - 10.4 = 10.0$. The value of F from Figure 10 is 0.98 for $\Delta S = 10.0$ at a wet bulb temperature of 75 F. The sizing index I is therefore $0.98 \times 10.0 = 9.8$. Figure 12 shows a value of $f_d a_e H_f$ of 45 for a face velocity of 400 fpm and a value of the sizing index I of 9.8. The value of H_f is then determined as follows:

$$\frac{f_d a_e H_f}{f_d a_e} = H_f = \frac{45}{12.7 \times 0.487} = 7.3 \text{ ft}$$

Example 2. A spray filled cooling tower operating with centrifugal nozzles at 7 psi under the conditions of rating of Table 6 is to cool under the following conditions: (1) temperature of water onto tower = 83 F; (2) temperature of water off tower = 77 F; (3) wet bulb of air entering tower = 70 F; (4) basic loading of water, $\overline{\text{GPM}_b} = 2.0$ gpm per sq ft; (5) vertical face velocity upward in spray chamber $v = 250$ fpm; (6) barometric pressure = 30 in. Hg. What is the height of the spray chamber required?

Solution: From the Basic Sizing Chart, (wet bulb temperature = 70 F) $S_{83} = 21.0$ and $S_{77} = 14.5$. The sizing scale intercept ΔS requires no correction at 70 F wet bulb; and the sizing index I is therefore, 21.0–14.5 = 6.5. From Figure 11, $f_d a_e H_f = 17.7$. From Table 6, for $H_f = 14$ ft, $f_d a_e = 1.3$; and $f_d a_e H_f = 18.2$. Therefore, the height of the spray chamber would be satisfactory at 14ft.

(Note:—If the air velocity were increased to 400 fpm, $f_d a_e H_f$ would be 14.8. From Table 6, $f_d a_e H_f$ for a spray chamber 10 feet high would be 14.7; and a height of spray chamber of 10 feet would be satisfactory if the air velocity were 400 fpm instead of 250 fpm.)

32. Variation in barometric pressure from the basic 30 in. Hg of the design curves for forced convection cooling towers may be considered in sizing by the use of certain rapid approximations. Reference to Equation 5 indicates that f_d decreases as V_a increases, a condition which exists when the velocity through the filling is held constant while the barometric pressure is decreased; and the variation in f_d is inversely as the 0.6 power variation of V_a. On the other hand, the variation in the driving forces of the sigma function between fixed temperatures increases inversely as a power less than one as the barometric pressure is decreased. Practically, one may consider that the increased driving forces of the sigma function at lower barometric pressures offset the decreased conductances at lower barometric pressures. Therefore, one should enter the design curves of Figures 11 through 15, correcting the actual air velocity by multiplying by the ratio of the actual barometric pressure to 30 in. Hg. For instance, with a barometric pressure of 25 in. Hg use the curves, with f_d computed by Equation 5 for 30 in. Hg. Enter the curves, however, with a corrected velocity for Figures 11 through 15 of $(400 \times 25)/30 = 333$ fpm.

In the case of spray towers, the principal conductance value is due to the relative projectile velocities of the drops with respect to the air. Therefore, the principal factors at increased altitudes are increased driving forces between given temperatures of water and wet bulb and decreased driving forces due to decreased air mass rate at a given air velocity. These latter factors may be estimated as balancing, and the curves may be used without further adjustment. Exact calculations for filled towers may be made by calculation of the conductances and driving forces for the exact conditions. In those cases, Equation 5 should be used with V_a evaluated for the elevation in question; and the driving force may be computed by means of the adjusted logarithmic[4,14f] mean applied to the actual psychrometric conditions of the problem.

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22. AUTOMATIC CONTROL DEVICES AND INSTRUMENTS

Part I. Control Devices

IN ORDER to utilize fully any form of refrigeration system, whether it be domestic, commercial, industrial, or air conditioning, it is desirable to have **automatic control** of temperature and of the equipment in the system. This can be accomplished by means of devices that are sensitive to these conditions and as a result thereof operate to start, stop, or regulate the source of cooling.

Automatic controls for refrigeration can be classified into three general types:

A. Temperature controls
    1. Room thermostats
    2. Remote-bulb controllers
B. Pressure controllers
C. Differential controllers
D. Refrigerant-flow controllers.

This chapter will deal with these controllers in the above order, covering their construction, function, and some application factors inherent in the various designs.

A. Temperature Controls

1. Room thermostats. A room thermostat may be defined as a temperature-sensitive device which is mounted entirely within the space it controls. It consists of five basic parts: (1) element; (2) adjustment means; (3) switching means; (4) a base upon which the first three items are mounted; and (5) a screen or cover to protect the above parts from physical harm and frequently to make the instrument more appealing to the eye. Quite often a thermometer is mounted on the screen and the adjustment means includes a scale.

Room thermostats may be classified by their **temperature-sensitive elements.** The earliest and perhaps the simplest element is made of bimetal, so named because of its fabrication from two dissimilar metals securely bonded into a duplex strip of flat metal. A change in temperature of the bimetal causes the inactive metal, usually Invar, an alloy having a very low thermal expansion coefficient, to change its length a relatively small amount while the active metal, brass or steel, measurably increases or decreases its length, depending upon the nature and amount of the temperature change. Since the two metals are bonded together, a force is exerted to relieve the strain imposed by the heat upon the bimetal. The element therefore deflects until the force developed by the heat applied to the element just balances the resistance to strain inherent in the metal. This power and deflection, although small, is usually used to operate an electric switch or air valve.

The bimetal element of a thermostat is frequently "U" shaped and mounted on a hinge attached to the thermostat base. One free end of the element rides on a cam or lever arrangement which is the **temperature-adjustment means.** The other free end either carries one of the electric contacts or operates a small **switching mechanism.**

Fig. 1 is a schematic diagram of a simple

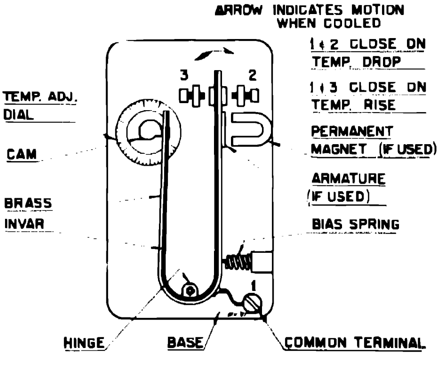
ARROW INDICATES MOTION WHEN COOLED
1 & 2 CLOSE ON TEMP. DROP
1 & 3 CLOSE ON TEMP. RISE
PERMANENT MAGNET (IF USED)
ARMATURE (IF USED)
BIAS SPRING
TEMP. ADJ. DIAL
CAM
BRASS
INVAR
3
2
1
HINGE
BASE
COMMON TERMINAL

Fig. 1. Simple Room Thermostat (SPDT Switching)

room thermostat having a SPDT (single-pole-double-throw) switching mechanism.

When a SPDT thermostat is used in a circuit requiring momentary closing of one circuit to start an operation and the opening of this circuit and momentary closing of a second circuit to stop the operation, the permanent magnet shown in the figure and its related armature are not used. If a SPDT thermostat is used to control separate loads, the magnet and armature are used to prevent contact chattering and sputtering with the resultant undesirable rapid starting and stopping of the equipment. This magnet is mounted in the path of the armature and it is the air gap between the two, when No. 2 contact is closed, that determines the differential of the unit; the smaller the air gap the larger the differential, and conversely.

A damper or valve motor is sometimes controlled by a thermostat of the former type and is moved in one direction by closure of one contact, in the opposite direction by closure of the other contact, and allowed to remain at rest whenever the movable contact is contacting neither stationary contact. Such a system is frequently called **"floating control."**

In some SPST (single-pole-single-throw) thermostats but one stationary contact is used and the magnet and armature are incorporated for reasons stated above. There are a few thermostats used in specific circuits that have no magnet and employ slow-moving contacts to produce very low differentials. These contacts are necessarily very limited in their load-carrying capacity.

In one special type of room thermostat, differential is introduced electrically by employing the start-stop-holding circuit principle. In this thermostat two contacts are closed or opened sequentially, the circuit requiring them both to be closed before the system can start and both to be open before the system can stop. Such a thermostat gives small operating differential but is also limited in its load-carrying ability.

"Manual differential" is the difference in **scale movement** that is necessary to cause starting and stopping of the equipment being controlled. "Operating differential" is the difference in **temperature** necessary to cause similar results. In a simple thermostat the latter is always greater than the former because of the inherent lag of the element plus the heat capacity of the base, screen, and other parts. Artificial heat is frequently applied internally in heating thermostats to increase the rate of response, which in effect reduces the operating differential.

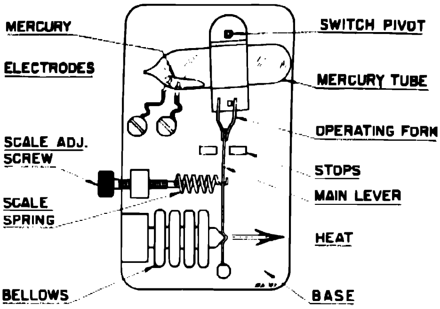

Fig. 2. Vapor-Filled Refrigeration Room Thermostat

A widely used element for refrigeration room thermostats is the **vapor-filled type.** In this type a metallic bellows is filled with a small quantity of a volatile liquid and the vapor pressure exerted on the bellows is used to operate electric switches or valves. **Fig. 2** is a schematic diagram of a thermostat of this type which employs a sealed contact structure.

The bellows of this instrument is evacuated and then filled with a small quantity of a liquid whose vapor pressure curve is within the desired temperature operating range. Freon, propane, ether, and other similar compounds are examples of fills used in these controls. A scale-adjusting spring is provided to determine the pressure against which the bellows must operate and consequently to determine the temperature at which the thermostat will control.

The main lever of the thermostat is pivoted at one point, the bellows and spring act at two other points, and at a fourth point is a power takeoff for operating the controlling member, either by electric contacts or a pneumatic valve. Differential is usually increased by means of an adjustable degree of lost motion in the drive mechanism.

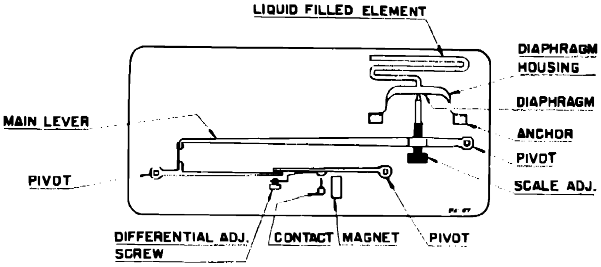

Fig. 3. Liquid-Filled Diaphragm Thermostat

If **open contacts** are used, they may be of the type described under the bimetallic type element, excepting that the detent may be accomplished by means of an over-center action rather than a permanent magnet. A popular switching mechanism is the so-called closed or **sealed contact** wherein two electrodes are sealed in a glass or metal envelope and the circuit is made by means of a small globule of mercury that flows back and forth within the envelope as it is tilted by the thermal element. Either type of contact structure can be had in SPST, SPDT, or special switching means.

Another type of sealed switch has a small globule of mercury in a sump in the glass enclosure. Hinged above it is the other electrode to which a small armature is fastened. A permanent magnet is then operated toward and away from the outside of the glass enclosure and consequently makes and breaks the circuit between the hinged electrode and mercury.

In operation the scale is set at the desired control point. An increase in temperature causes an increase in internal pressure which when it is sufficient overcomes the force of the spring and operates the main lever to tilt the switch, closing its contacts. A drop in temperature causes the converse to happen.

Another very popular switching means for this type thermostat consists of a small **"micro" switch** having either an over-center snap action or a magnetic detent. The advantage of this switch and also the sealed mercury switch is their ease of replacement following failure in the field. The micro switch is usually available in SPST or SPDT switching action.

Generally speaking, this type of thermostat may handle fairly large electrical loads directly. Units have been manufactured which can handle 45 amps at 115 volts. Also, this type of thermostat, because of its greater mass and the larger motion required to operate the contacts, will have a greater operating differential than the bimetallic types. If sealed contacts are employed, they will naturally require no cleaning during their life and are unaffected by moisture or other corrosive atmospheres. If the mercury switches are used, the controls must be accurately levelled as a small degree of tilt will throw the instrument off calibration or in some cases will cause it to short-cycle.

A third common type of element used for room thermostats is a relatively new one, consisting of a **hydraulic fill system** wherein the expansion and contraction of a liquid in a closed system caused by temperature changes is used to operate a metallic diaphragm which in turn operates a switch or a valve. **Fig. 3** is a schematic diagram of this type of thermostat.

The element usually consists of a short length of small diameter tubing, one end of which is fastened to a diaphragm housing so that the inside of the housing and the inside of the tubing are connected. The diaphragm is then securely fastened inside the housing, leaving very little space between the housing and diaphragm. After the diaphragm is put in place, the system is evacuated and filled with a liquid having the desired characteristics and the free end of the tubing is sealed. A lever system designed to multiply the diaphragm motion greatly is provided with an adjustment to vary the position of the diaphragm at which it moves the contacts.

Operating differential is increased by an adjustable spacing in the lever system. The available motion in this type of element is quite small but the force developed is large. Because of this last characteristic it is absolutely essential that the diaphragm housing and tubing be of sturdy construction. It is also imperative that the diaphragm be sufficiently rigid to insure its motion occurring at the point of contact between it and the push-rod, and not at other points in its area. This last requirement is quite often accomplished by means of ribbing the diaphragm radially from the center. These elements can often be damaged beyond repair by denting the tubing or housing, which causes a reduction in the internal volume of the system.

Because of the small motion and tremendous force that is available, these units usually operate through a tremendous leverage before actually driving the contacts. In spite of the large forces developed by a liquid fill the actual work possible from it is less than with other control elements, unless the bulb is made extremely large. This is due to the extremely small motion produced for a given temperature change.

2. **A remote-bulb temperature controller** may be defined as a temperature-sensitive device in which the element or bulb is placed at the point in space whose temperature is to be controlled and the remainder of the instrument is placed elsewhere in the space or in a separate room. This type of controller consists of: (1) remote bulb and its capillary tubing; (2) case; (3) bellows or diaphragm; (4) scale and differential adjustment means; and (5) switching or valve-actuating means.

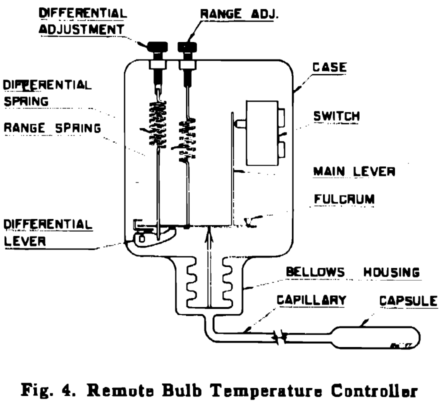

Fig. 4. Remote Bulb Temperature Controller

Here again the best division of types of remote-bulb temperature controls appears to be by types of sensitive elements. The most common type uses the **vapor-tension fill** whose basic operating principle has been earlier described. **Fig. 4** is a schematic diagram of a controller of this type.

Since these controllers are seldom used in public places or rooms where appearance is important, they may be less attractive than the room thermostat, and have a more functional appearance. Methods of operation, types of switching means available, and structural features of this unit are similar to those given under room thermostats of the same type with two basic exceptions.

First, the bellows, instead of being the sensitive element and power source, as in a room thermostat, becomes the power source only. The temperature-sensitive element is in the remote bulb as within it is contained the active liquefied portion of the vapor fill. The remote bulb is connected to the bellows by means of flexible capillary tubing or by means of long rigid tubing.

Secondly, the differential is usually increased by means of a second spring which adds additional loading to the main lever through a differential lever which engages the main lever for a portion of the time when the main lever is in position to operate the switch. If adjustable differential is not required this mechanism is omitted from the instrument.

Even with a small bulb, which permits good control sensitivity, the work available from a vapor-filled element is greater than with the other types of elements, being limited only by the size and type of bellows.

Fig. 5 illustrates three common types of vapor-tension fills employed in bellows assemblies. On the left is a so-called fade-out or **limited fill,** which is employed for low-temperature applications in which the bulb is always at a lower temperature than the bellows. Because it is at a lower temperature, the liquid in the system will always exist in the bulb and nowhere else, and since it is the temperature of this liquid surface which determines the pressure in the rest of the system, proper control obtains from variations in bulb temperature. Such bulbs are of small size, relatively smaller than illustrated.

The right-hand illustration in Fig. 5 shows the so-called **high-temperature fill** in which the bulb is always at a higher temperature than the bellows. Note that in this diagram there is enough liquid in the charged system to fill the bellows and capillary completely and to fill the bulb partially. Thus even though the liquid is cooler in the bellows than in the bulb, there will always be a liquid surface in the bulb which controls the pressure in the rest of the system, and the remaining liquid merely becomes a hydraulic pressure transmitting means. Thus the bulb will always be the controlling point as long as it is warmer than the bellows.

The center shows a common **cross-ambient control arrangement** in which a larger bulb is employed, sufficiently large so that if the bulb is colder than the bellows the entire charge is contained in the bulb, and the bulb becomes the controlling point of the system. If, on the other hand, the bellows is colder than the bulb, it fills with liquid, as does the capillary, and yet due to the size of the bulb it retains sufficient liquid to do the controlling. Thus, regardless of whether the bellows temperature is above or below the bulb temperature, the desired control point is obtained at the bulb. This distinction is very important in all types of work where the ambient temperature at the bellows or capillary varies from above to below the setting at which the bulb must operate the control.

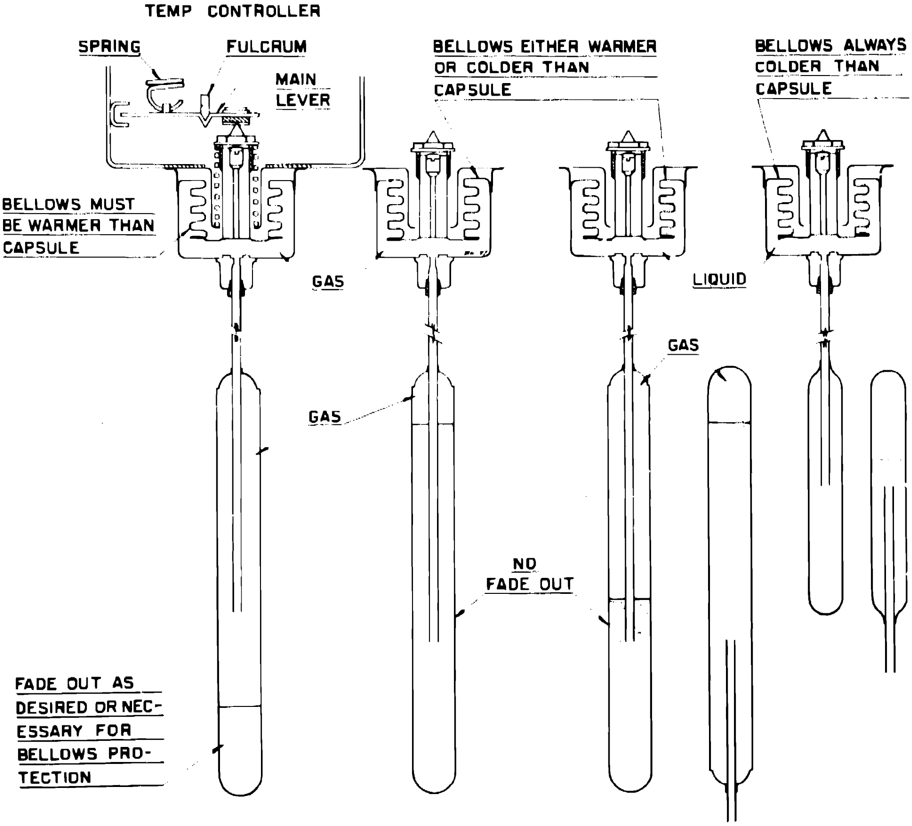
TEMP CONTROLLER
SPRING
FULCRUM
MAIN LEVER
BELLOWS MUST BE WARMER THAN CAPSULE
GAS
GAS
FADE OUT AS DESIRED OR NECESSARY FOR BELLOWS PROTECTION
BELLOWS EITHER WARMER OR COLDER THAN CAPSULE
NO FADE OUT
LIQUID
GAS
BELLOWS ALWAYS COLDER THAN CAPSULE

Fig. 5. Filling Arrangements for Bellows Assemblies

Another type of remote bulb controller is the liquid-filled type which in outward appearance can resemble the room thermostat with liquid-fill element earlier described. The active element, as the name applies, consists of a length of capillary tubing that extends out of the case of the instrument instead of being coiled up within it and usually terminates in an enlarged bulb. This element can therefore be placed at the position in space requiring the temperature control and the case can be mounted elsewhere, even in a separate room if desired.

B. Pressure Controllers

Pressure controls for refrigeration may also be classified by means of their power elements. There are three types now in general use: (1) bellows; (2) diaphragm; (3) bourdon tube.

The **bellows type** pressure control is identical in construction to the remote-bulb temperature control except that the pressure connection to the power element is made by connections such as pipe threads or flare connections, or by a short length of capillary enlarged on the free end to a standard tube size.

Its internal operation is also identical with that of the temperature controller, since the bellows is now subjected to refrigerant pressure and not the pressure developed by the fill. In order for the bellows type control to operate in the subatmospheric range, it is necessary to incorporate a compression spring within the bellows housing. This spring is in addition to the scale adjustment spring, but its force is opposite to that of the latter.

The force available for doing work in a control of this type is dependent first upon the pressure in the system and secondly upon the area of the bellows. If the bellows is large enough, a tremendous force can be had for operating heavy switches, water valves, or even refrigerant valves. In these heavy-duty controls the minimum differential is necessarily quite large because of the spring rate of the bellows and opposing spring system.

A second type of element in which the available motion is much less than the bellows type is the **diaphragm type.** Since the motion is very limited, large leverages are needed to produce motion adequate to operate most switches. In this type of control the leverage and spring system is quite similar to those earlier described. It is essential, however, that the diaphragm be restrained either by a backing plate or reinforced with radial ribs in order to guarantee the motion occurring at the point of contact between diaphragm and push rod.

The third type of element is the **bourdon tube** type used for many years in pressure gages. **Fig. 6** is a schematic diagram of this type.

As shown by the cross-sectional view of the element, it consists of a flattened tube rigidly held at one end and free to move toward or away from the free end by means of the scale adjustment screw. It is the position of the switch relative to the end of the bourbon tube that determines the scale setting. Differential can be increased by employing a linkage wherein the lost motion between bourdon tube movement and switch is increased.

C. Differential Controllers

Controllers used to maintain a given difference in pressure or temperature between two pipe lines or spaces, are known as differential controllers. These controllers, since they must sense conditions in two different locations, must have two elements, either pressure or temperature-sensitive.

Fig. 7 is a schematic diagram of a **differential-pressure** controller, using bellows for elements.

As shown, the two elements are rigidly connected by means of a rod so that motion of one causes motion of the other. On the connecting rod is a power takeoff that operates contacts or valves (SPDT contacts are shown). A compression spring has

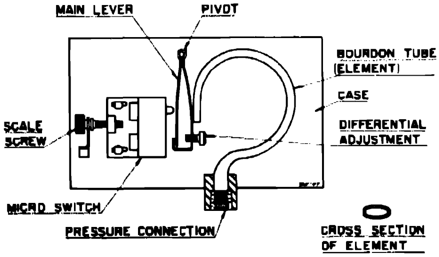

Fig. 6. Bourdon Tube Pressure Control

been provided to permit setting of the differential pressure at which the device operates. The sum of the force developed by the low-pressure bellows and the scale spring equals the force developed by the high-pressure bellows at the control point.

In discussing this type of controller, two terms must be defined and thoroughly understood. **Instrument differential** is the difference in the pressure or temperature that is maintained between the high and low-temperature bulbs of the system. **Operating differential** is the difference in pressure or temperature that is necessary to operate the switch at the instrument differential that is to be maintained. Operating differential can be increased by means of a second spring that acts in the same direction as the first and which takes effect within the operating differential of the instrument. A second method could be adjustment of the distance between collars Z-Z on the connecting rod; the farther these are apart the larger the operating differential.

The foregoing discussion on differential controllers has dealt entirely with the pressure type. The **temperature-sensitive** type is made by merely replacing the pressure bellows with vapor tension elements, whose theory of operation has been earlier described. Where a constant instrument differential must be maintained throughout a large band of temperatures, it is usually necessary to use a different fill in one element than in the other if of the vapor type, or to use elements and drive of the liquid-filled type.

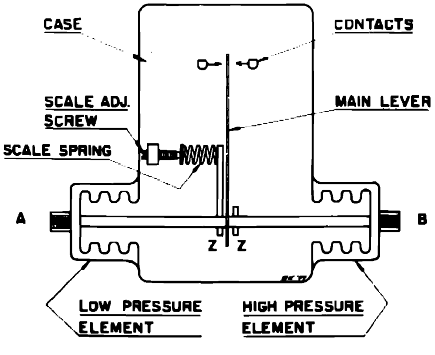

Fig. 7. Differential-Pressure Controller

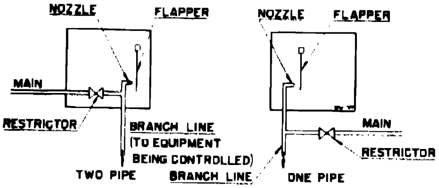

Fig. 8. Pneumatic Controllers

A second type of differential-temperature controller employs two capsules and capillaries connected to one bellows and a liquid fill. This is known as a **constant-volume** type fill, since the operating point corresponds to a constant volume of the two capsules, capillaries, and bellows. It should be noted that if the two bulbs have equal volume, a rise in the temperature of one bulb requires an equivalent fall in the temperature of the other to maintain the operating point.

The foregoing discussion on controls has dealt entirely with those having electric contacts to do the actual operating of controlled equipment. Another very popular type uses a small air valve, and positions dampers or valves by means of regulating the air pressure on the bellows or diaphragm of the equipment being controlled. These are known as **pneumatic controllers** and can be classified as either one-pipe or two-pipe devices. **Fig. 8** shows the difference between these.

In these controls compressed air at about 15 psi pressure is used as the activating medium and is supplied by the main to the controller. In the two-pipe system, the restrictor is built into the controller in the main to the nozzle, and controls the rate at which air is supplied to the nozzle and consequently the pressure in the branch line. If the flapper completely seals the nozzle, full main pressure is built up in the branch line, and if the nozzle is not sealed at all by the flapper, no pressure is built up in the branch line, as the air leaks out the nozzle faster than it can be supplied through the restrictor in the main. In the one-pipe system the restrictor is external and inserted in the main ahead of the tee that goes to the controller and controlled equipment.

To understand automatic controls thoroughly and use them intelligently, some knowledge of their applications, limitations, and operating characteristics is essential.

Effects of altitude changes. Any control that relies on pressure to operate a switch or valve is subject to variations in operating point as the atmospheric pressure changes. Normal variations in atmospheric pressure do not noticeably change the operating point, but a change in altitude will affect the control point by an amount governed by the change in absolute pressure. The greater the altitude, the lower will be the operating point of a conventional pressure or temperature control. If a pressure control is being considered, the change is directly proportional to the change in altitude. If a temperature control is being considered, the change in calibration equals the change in vapor pressure translated to degrees temperature, at the scale setting being used. These characteristics are especially important when controls are being selected for use in planes, buses, and trains that in normal use are subjected to wide variations in altitude.

Bimetallic and liquid elements and special bellows-type elements using a compensating bellows are not affected by changes in altitude, so if altitude is the only problem to overcome, their use for these applications should be considered.

Mobile uses of controls. In discussing mobile applications, it must be remembered that there is always vibration present which will affect the operation of some types of controls. The types that are wholly unsatisfactory for mobile applications are those employing mercury tubes, slow-moving contacts, or any that are subject to leveling. There remains, then, only those controls that have positive contact action and employ mechanical or magnetic detents on their contacts. Generally speaking, the so-called micro-switches are also satisfactory for this application.

Levelling of controls. Any control that employs a mercury switch is extremely sensitive to leveling and should be mounted so that the sides of the base are vertical or the top is absolutely level. Great care is taken at the factory to calibrate the instrument in the correct position, and in order to use its accuracy fully, it must be properly installed. Some remote-bulb and pressure controllers are conveniently equipped with a plumb-bob to facilitate proper installation. In some instances off-level installation will not only upset the calibration but will also render the instrument completely inoperative.

Effect of internal heat. All room thermostats employing contacts are to some degree affected by the electrical load being handled. This is because of the electrical resistance of the current carrying parts generating heat when current is flowing. In refrigeration thermostats, this small amount of heat tends to increase the operating differential, a seldom desirable characteristic. A common name for this phenomenon is "**contact heat.**" It is probably most noticeable in bimetallic-type controls, because the element itself frequently carries current and the contacts are usually fastened to the element. Vapor-tension and liquid-filled thermostats are somewhat affected but usually to a smaller degree, because the heat generated by the load-carrying parts reaches the element mainly by convection and not by thermal conductivity. Thermostats employing micro-switches can be designed to minimize greatly the effect of contact heat by isolating the switch from the element with material having low thermal conductivity. As is obvious in a given thermostat, the contact heat increases as the square of the contact load in amperes and consequently the greater the load the greater the operating differential.

Thermostat location. Since any temperature control has inherent in it some thermal lag, its performance is affected by the rate of temperature change and rate of fluid motion to which the element is subjected. The operating differential varies directly as the rate of temperature change and inversely as the rate of fluid motion across the element. Proper selection of element location can often minimize the effect of these two variables by subjecting the control to average and not extreme conditions. For example, it would probably be undesirable to mount a room thermostat where it would get the immediate full blast of cooled or heated air. Mounting of the capsule of a remote-bulb controller in a duct having a high air velocity will usually produce closer control than if it is mounted in a duct having a lower air velocity. In some remote cases it might be desirable to use a small blower to circulate air over the element.

Some unique designs have been marketed to minimize these effects. One remote-bulb control has been developed which includes fins on the thermal bulb; a thermostat has been marketed which includes a shunt type of heater to reduce operating differential. The value of the former is questionable as the mass added by the fins almost offsets the gain made by increased element area. The latter method is entirely practical since it requires merely a high-resistance heater across the switch contacts which generates heat when the unit is off. This heater merely accelerates the element action back to an "on" position of the contacts. The heat generated by the heater affects the element in the same manner as the increase in space temperature resulting from the equipment being inoperative.

Explosive atmospheres. An important consideration when selecting controls for an operating room is the explosiveness of the atmosphere that is present during operations. So-called "sealed-in-glass" contacts are seldom considered explosion-proof, and other methods must be used to eliminate any possible electric spark within the atmosphere. Quite frequently an explosion-proof case can be used to surround the control case and contacts, permitting only the capsule and capillary to extend within the conditioned space. It is frequently possible to use a long capillary tubing and mount the instrument case in an explosion-free atmosphere elsewhere in the building.

Cross-ambient effects. All remote thermal elements are to some degree affected by the temperature of the operating means, whether bellows or diaphragm, and by the temperature of the capillary that transmits pressure to it. The degree and type of effect is dependent upon the type of fill, which is illustrated in **Fig. 9.** In the examples shown it is assumed that when the capillary and the head of the instrument are at 80 F, the instrument is on scale.

In a liquid-filled element, a decrease in the ambient temperature surrounding the aforementioned parts creates a gradual and continuous increase in the actual control point. Increasing the ambient temperature will cause the converse to happen. The degree of this change depends on the relative volume of the element subject to the controlled and to the ambient temperature.

In a vapor-tension fill of the cross-ambient type, the actual control point coincides with the desired control point except for a small span near the temperature at which the ambient temperature and scale setting are equal. Within this span the differential is apt to be increased and erratic operation may be experienced. This phenomenon is caused by the transfer of the liquid back and forth between bellows and capsule.

In the fadeout or limited type of vapor-tension fill, the actual control point coincides with the desired control point as long as the ambient temperature surrounding the head and capillary of the instrument is above the capsule or bulb temperature. When the former falls below the latter, complete loss of control will result as the small droplet of active fill leaves the capsule and enters the bellows housing; actually, then, the bellows becomes the temperature-sensitive part of the thermal system.

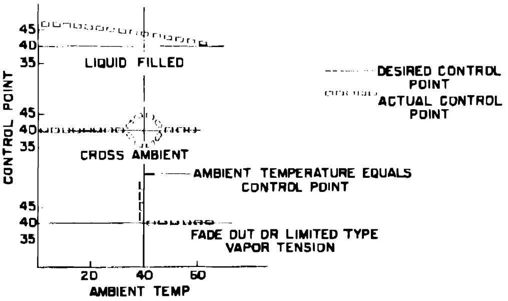

Fig. 9. Effect of Ambient Temperature on Controllers

Limit controls. A further and most important factor to consider when selecting automatic controls is the type of operation needed, controlling or high-limit (safety). In the former of these, quite frequent operation of the control mechanism occurs and any failure of the control is likely to be noticed before damage is done. In the latter the control may go for days, months, or possibly years without operation, and then

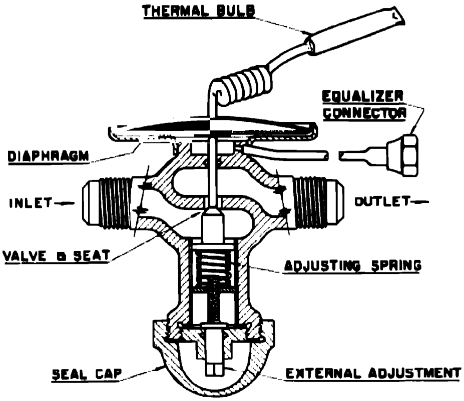

Fig. 10. Thermostatic Expansion Valve

because of some unsafe condition, which the control was installed to protect against, it will be required to operate immediately. At this time operation is a "must," or serious damage to equipment or even property might result.

D. Refrigerant-Flow Controllers

Expansion valves. One of the most important control functions within the refrigerating system is that of the admission of the refrigerant to the evaporator. An expansion valve is normally employed for this purpose except in the very smallest systems. Expansion valves fall into two general classes; (1) those which are operated by changes in the level of the liquid refrigerant in a chamber; and (2) those which are actuated by changes in pressure. This second class of valves has many of the same control characteristics that exist in other automatic control devices and will therefore be discussed in some detail.

Thermostatic expansion valves. A typical thermostatic expansion valve is shown in **Fig. 10.** It consists of a valve body containing the valve, which is positioned by the diaphragm and the adjusting spring. The position of the diaphragm depends on the difference in pressure above and below it and the force of the adjusting spring. The space above the diaphragm usually contains the same refrigerant which is used in the refrigeration system, and the pressure therein is determined by the temperature of the thermal bulb which is inserted in or strapped to the suction line of the evaporator being controlled. The pressure below the diaphragm is made the same as the pressure within the evaporator by means of the equalizer connection.

The amount of refrigerant which a thermostatic expansion valve allows to enter the evaporator will always be such as to keep the difference in pressure above and below the diaphragm equal to the force exerted by the adjusting spring. Since the pressure within the thermal bulb can be greater than the pressure in the equalizer connection only in an amount proportional to the superheat of the refrigerant in the suction line, these valves are sometimes referred to as constant superheat valves.

The action of the valve is to fill the evaporator with refrigerant containing some portion of unevaporated liquid, except for the relatively small distance near the suction side which is required to superheat the gas. Thus the evaporator is operated at high efficiency over a large range of load conditions.

For the best control with a thermostatic expansion valve, and the best use of the evaporator, the superheating region should be in contact with the warmest portion of the air or liquid being cooled. Where this is not possible or where the difference in temperature between refrigerant and the air or liquid being cooled is extremely small, a separate heat exchanger can be used to produce the superheat to operate the valve.

Where rapid changes in evaporator temperature or pressure are likely to occur some hunting may be experienced with thermostatic expansion valves. Hunting usually can be reduced either by slightly insulating the thermal bulb from close contact with the suction line or by the use of stiffer adjusting springs.

Automatic expansion valves. The automatic expansion valve shown in **Fig. 11** has its valve positioned only by changes in pressure in the evaporator. The pressure in the inlet side of the evaporator is transmitted through the pressure equalizer hole to the space below the diaphragm, and an increase in evaporator pressure tends to close the valve. The pressure under the diaphragm is opposed by the adjusting spring. The adjusting nut allows the selection of the evaporator operating pressure.

The diaphragm is usually subjected to the same temperature that exists in the evaporator, and moisture from the air must be prevented from coming in contact with the diaphragm by the flexible seal cap. In order for the automatic expansion valve to operate at pressures within the vacuum range, it must contain a vacuum spring in order to close the valve against atmospheric pressure.

The automatic expansion valve is used wherever it is desired to have a refrigeration system or evaporator operate at constant load. It should not be used where loads much lighter than normal might occur, since this would result in too great an amount of liquid entering the evaporator.

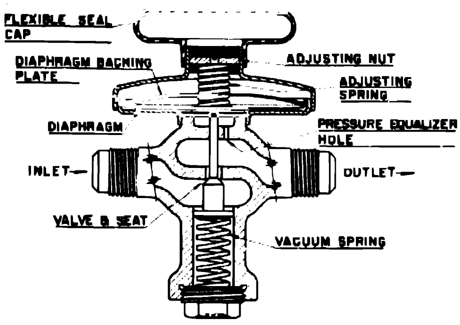

Fig. 11. Automatic Expansion Valve

The non-evaporated portion would be returned to the compressor as liquid and might damage the compressor.

Part II. Instruments*

THIS section deals with instruments used for test and other measurement purposes. The instruments described are primarily for indicating but some of them may also be used for recording. The section is intended to serve as a guide in the selection of measurement equipment. More complete details for the use of the instruments are available in the references.

Temperature Measurement

Table 1 shows the principal types of instruments used for temperature measurement, their applications, range, order of accuracy and limitations. Principally used in the ordinary temperature range are glass-stemmed thermometers, thermocouples and resistance thermometers, in that order.

While they are direct-reading and simple to use, **glass-stemmed thermometers** are subject to corrections for initial graduation, aging, pressure and depth of immersion.

Thermocouples are especially well suited for remote reading and of a number of stations. They are ideally suited for measurement of temperature of solids as well as of fluids. In a fluid circuit such as a refrigerating system, it is possible by the attachment of thermocouples on the surface of the pipe to determine the approximate temperature of the fluid within. Insulation of the pipe improves the accuracy of this method. In order to develop their full possible accuracy, thermocouples should be used with a potentiometer equipped with a standard cell and a cold junction held at constant temperature (or the equivalent electrical compensation). They should also use carefully standardized and calibrated wire. They may be used with lesser accuracy with a microammeter and high resistance in place of the potentiometer.

Thermocouples may be used in parallel to indicate the average temperature but it is important for this purpose that the wires be carefully equalized in resistance. They may also be used in series for the same purpose without the necessity of equalizing resistance. They are also sometimes used in series as a thermopile in order to increase the output voltage and to simplify the problem of measurement. Many times it is convenient to arrange the thermocouples for differential temperature measurements in such instruments as a pyrheliometer, heat-flow meter, etc.

Resistance thermometers are of especial value for remote readings of gases requiring a high degree of accuracy. They are useful where a grid of resistance wires can be used to indicate average temperature of a stream.

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* This part of Chapter 22 was prepared by C. M. Ashley.

Table 1. Measurement of Temperature

No.	Measurement Means	Application	Range, F	Order of accuracy, F	Limitations
1	Glass-stem thermometers				
	Mercury-glass thermometer	Temp of gases and liquids by contact	−38/575	Less than 0.1 to 10	In gases, accuracy affected by radiation
	Alcohol-glass thermometer	"	−100/100	"	"
	Pentane-glass thermometers	"	−200/70	"	"
	Jena or quartz mercury nitrogen thermometers	"	−38/1,000	"	"
2	Gas thermometer	Primary standard	−459/1,000	Less than .01	Requires considerable skill to use
3	Resistance thermometers				
	Platinum-resistance thermometer	Precision; remote readings; temp of fluids or solids by contact	−320/1,800	Less than .02 to 5	High cost; accuracy affected by radiation in gases
	Nickel-resistance thermometer	Remote readings; temp by contact	−150/300	.3	Accuracy affected by radiation in gases
4	Thermocouples				
	Pt–Pt-Rh thermocouple	Standard for thermocouples	500/3,000	.1 to 5	High cost; also, requires expensive measuring device
	Chromel-alumel thermometer	General testing of high temp; remote rapid readings by direct contact	Up to 2,200	0.1 to 15	Less accurate than above
	Iron-constantan thermometer	"	Up to 1,500	0.1 to 15	Subject to oxidation
	Copper constantan thermometer	Same as above, especially suited for low temp	Up to 600	0.1 to 15	
5	Beckman thermometers (metastatic)	For differential temp in same applications as in glass stem thermometer	5 C diff	0.01 C	Must be set for temp to be measured
6	Bimetallic thermometers	For approx temp	0/1,000	1, usually much more	Time lag; unsuitable for remote use; unreliable
7	Pressure-bulb thermometers				
	Gas-filled bulb	Remote-testing	−200/1,000	2	Caution must be exercised so that installation is correct
	Vapor-filled bulb	"	20/500	2	"
	Liquid-filled bulb	"	− 50/1,200	2	"

Table 1. Measurement of Temperature (Concluded)

No.	Measurement Means	Application	Range, F	Order of accuracy, F	Limitations
8	Optical pyrometers	For intensity of narrow spectral band of high temp radiation (remote)	1,500 upward	15	
9	Radiation pyrometers	For intensity of total high temp radiation (remote)	Any range		
10	Seger cones (fusion pyrometers)	Approx temp (within temp source)	1,000/3,600	50	
11	Melting and boiling points of materials	Standards	All except extremely high temp	Extremely precise	For laboratory use only

Radiation effects. Gas temperature readings may be affected by radiation from surrounding surfaces at different temperature, particularly at high temperature levels. Glass-stemmed thermometers are particularly affected, due to the large bulb with correspondingly low convection transfer and to the high emissivity of glass. The use of thermocouples having a minimum wire size is preferred where radiation is a problem. The radiation effect may be minimized by applying directly to the couple a coating of low-emissivity metal such as aluminum, silver or platinum. Surrounding shields of low-emissivity metal are helpful where the gas velocity is appreciable. Complete correction of the radiation effect may be obtained by: (1) using an aspirating tube around the couple, reading temperature at a series of gas velocities and projecting to infinite velocity; (2) using a series of thermocouples of different sizes and projecting readings to zero diameter; (3) using two couples of known and different emissivity; and (4) using heated shields.

Sampling and averaging. Temperature is usually a point measurement, but it is frequently assumed representative of the value over an area as on a surface or in a flowing stream. However, such an assumption is rarely safe without an exploration. Where good accuracy is desired in the measurement of average temperature as of a stream, it is desirable to divide the stream cross-section down into a number of equal parts and measure the temperature at the center of each. Even the average of these readings may not be right if the velocity is not also uniform. While this can be compensated by multiplying the temperature by the velocity at each point and averaging, the more desirable solution is to find means of minimizing the variation. In drawing air from a room, the temperature variation can be minimized by drawing all of the air from the same level. In measuring temperature in a duct or pipe following a heat transfer element, it is desirable to use a mixer. While mixture may be obtained by turbulence alone, a method many times more effective is by transposition—the stream is divided down into parts and the alternate parts passed through each other. Mixers of this sort may be arranged to re-mix fluid having almost any type of stratification.

Thermometer wells. Sometimes it is necessary to use thermometer wells in the measurement of fluid temperatures. The well should be made to fit the thermometer as snugly as possible and should be filled with a high conductivity liquid (oil, water or mercury if suitable) in order to cut conduction losses to a minimum. The well should be as long and thin-walled as practicable. The pipe adjacent to and including the well should be well insulated to prevent temperature difference which would affect the accuracy of the reading.

Effect of velocity. When thermometers are placed in a moving stream of fluid, the temperature indicated reflects a part of the energy of motion of the stream, usually in the range of 60 to 80%. At ordinary velocities this factor is negligible, but at higher velocities it becomes extremely important, and under these conditions it is necessary to design special temperature-measuring instruments.

Measurement of Relative Humidity

Table 2 shows the principal types of instruments used for the measurement of relative humidity.

Relative humidity is normally measured by the use of a **wet** and **dry bulb thermometer,** the humidity being indicated on a psychrometric chart. Good accuracy requires the use of velocities of 1,000 to 1,500 fpm over the bulbs. In a room this is normally obtained by swinging a sling psychrometer by hand or better by means of an aspirating-type psychrometer. In a duct the air velocity is usually of the right order. If not, the reading can be taken in a special sampling duct which is adjusted to the proper velocity, or a velocity correction can be applied to the reading.

The wet bulb surrounded by a wick is wet with water close to the wet bulb temperature and readings taken at frequent intervals until the temperature is stabilized. At high wet bulb depressions the wet bulb temperature may not be reached before the wick starts drying out. In this case the air can be measured as to temperature, drawn through a heat exchanger for cooling, the wet and dry bulb temperatures measured and the original conditions calculated.

Below 32 F a thin film of ice may be frozen directly on the bulb. Because of the small wet bulb depression in this region, great care must be taken in its measurement. The psychrometric chart over ice is to be used.

At sub-zero temperatures, **other means** must be used. Among these are: a cell whose electrical conductivity varies with humidity; a cell containing hygroscopic salt which is heated electrically to indicate dew point; a compressed air stream expanded adiabatically to the fog point; and dew point indicators and hygrometers. These methods used for sub-zero temperatures are in general also usable in the normal temperature range, and the first two are particularly well adapted for recording.

Hygrometers are widely used for approximate indication of relative humidity for temperatures up to over 100 F. They are actuated by hygroscopic elements (usually hair). Their accuracy tends to be impaired by exposure to very high and low relative humidities and temperatures over 125 F. Periodic recalibration is desirable.

Pressure Measurement

Absolute pressure. Instruments for measuring absolute pressure are shown in **Table 3.**

Very low absolute pressures (25 to 5,000 microns of mercury) may be measured by means of the McLeod gage, the Pirani gage and electronic discharge types. In the **McLeod gage,** a known volume of gas is trapped and raised to a predetermined pressure at which its remaining volume indicates the initial pressure. While the accuracy of measurement is good, the instrument is not direct-reading since it must be carried through a series of operations to obtain the reading. Furthermore, it is not well suited to the measurement of total pressure where appreciable quantities of condensible vapors are present, since these tend to condense out during the compression of the gas. The **Pirani gage** measures the thermal conductivity of the gas; thus the accuracy of the measurement depends upon the knowledge of the composition of the gas whose pressure is being de-

Table 2. Measurement of Humidity

No.	Measurement Means	Application	Range, F	Order of accuracy, F	Limitations
1	Wet bulb thermometer with dry bulb thermometer	Room of building, outside air, air moving in ducts; standard method	0 to 500	.1 to 1	Should be used in air stream moving 1,000 to 1,500 fpm or correction made. Unsuitable for recording at sub-freezing temp
2	Dew point apparatus				
	Condensation type	Occasional laboratory work; not widely used	−180 to 200	2	Cumbersome, expensive, difficult to use with precision; unsuitable for remote use
	Fog type	Simple wide range method	−100 to +100	2	Series of readings necessary for determination
	Electrical conductivity type	Accurate measurement of dew point; suitable for remote use	0 to 200	1	Unusable below −20 F and below 15% rh or for any moderate humidity at temp under 0 F
3	Hygrometer (for relative humidity)				
	Hair gyrometer	For direct rh in air where motion is slight	0 to 100% rh −40 to 150 F	3% rh	Considerable lag; low sensitivity; adversely affected by temp above 125 F and rh below 20%; frequent calibration required when used at extremes of range
	Hygroscopic material other than hair	Same as above—also for high sensitivity	0 to 100% rh −40 to 150 F	2 to 5% rh	Frequent calibration required when used at extremes of range
	Electrical conductivity type	For remote use	"		Requires frequent calibration
4	Infra-red radiation	While usable at all temperatures it is uniquely applicable to extremely low temp			Requires the use of very costly equipment
5	Standard salt	For standards			
6	Chemical analysis				
7	Absorption and weighing				Cumbersome process

termined. Other gages working in this pressure range use a discharge of electrified particles impinging on a screen.

For absolute-pressure measurements in the range of 0.1 to 10 in. of mercury, an **absolute-pressure manometer** is a preferred method of measurement. Unfortunately, most of these instruments are not readily portable. In a somewhat higher pressure range (0.5 to 30 in. Hg), a **differential mercury manometer** is frequently used to measure gage pressure and corrected to absolute pressure by the use of a **barometer.** Barometers of the mercury manometer type are special absolute-pressure manometers designed for the measurement of atmospheric pressure. For good accuracy, manometers and barometers must be corrected for temperature.

In refrigerating systems, the pressure of the high or low-side is frequently determined by the measurement of the temperature of condensing or evaporating refrigerant. In obtaining this measurement, care must be taken to make certain that there is neither sub-cooling, superheat or non-condensible gas present.

Differential pressure. Instruments for measuring differential pressure are shown in **Table 4.**

Very low pressure differential can be measured by means of a **micro-manometer.** One form of this is a hook gage with micrometer adjustment and magnifying-means to increase the accuracy, distinctly a laboratory instrument.

For somewhat higher pressures in the range of 0 to 10 in. of water, an inclined **draft gage,** as illustrated in **Fig. 12,** is commonly used. While this instrument must be filled properly and leveled, it is generally suitable for use either in the laboratory or in the field, and is commonly used for air velocity measurements together with the **pitot tube.**

For higher pressure readings where good accuracy is required, a "U" or **well-type manometer** is available. It is generally used either with water or mercury as a working fluid. When a mercury manometer is used for the measurement of liquid pressure difference, care must be taken to see that the connecting lines are filled with liquid and correction for the liquid density and for temperature must be made.

For the measurement of medium to high pressures, **bourdon tube gages** are normally used. These are available in a wide variety of ranges and accuracy. They are quite subject to damage due to vibration, pressure pulsation or excess pressure, and for accurate work must be calibrated frequently against a standard, such as on a dead-weight tester.

Velocity Measurements

Instruments for measurement of velocity are shown in **Table 5.**

No completely satisfactory instrument is available for measurement of low air velocity (0–100 fpm). Several types of **hot-wire anemometers** have been designed, both directional and non-directional. Unfortunately, the accuracy of many of these is questionable in the low end of the range and they should, therefore, be used with great caution. A **swinging-vane type** anemometer is also available for use in this range.

Another useful device for very low velocities is to time the rate of movement of a smoke puff, feather or piece of lint. The **Kata thermometer** uses a very large bulb which is heated and then allowed to cool between two set temperatures, the rate of cooling in conjunction with the ambient temperature indicating the velocity. Unfortunately, it is subject to several errors and can be considered as approximate only. While the above methods are well suited to measurement of velocities in an open

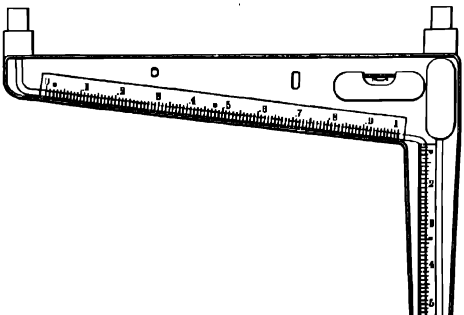

Fig. 12. Inclined Draft Gage with High-Range Leg

Table 3. Measurement of Absolute Pressure

No.	Measurement Means	Application	Range	Order of accuracy	Limitations
1	McLeod gage	Very low abs press	0–0.1 in. Hg	2–5%	Not direct reading; vapors tend to condense out
2	Pirani gage	"	"	2–5%	Must be calibrated for gas composition on which used
3	Absolute-pressure manometer	Moderately low abs press	0–30 in. Hg or above	0.01 in.	Not readily portable
4	Barometer, mercury—manometer type	Atmospheric press		0.001–0.01 in.	"
5	Differential manometer and barometer	Moderately low abs press	0–60 in. Hg or above	0.01 in.	Awkward observation; requires two readings
6	Evaporating liquid temp	On refrig machine and for use during evacuation	Depends on liquid	1%	Liquid must be kept supplied; can only be used where liquid evap will not damage system

Table 4. Measurement of Differential Pressures (Gage)

No.	Measurement Means	Application	Range	Order of accuracy	Limitations
1	Micromanometer	Very low press diff	0.000–6 in. H_2O	0.0005 in. H_2O	Not readily portable; not easy to use with pulsating pressure
2	Draft gages	Moderately low press diff	0.0–10 in. H_2O	0.005 in.– 0.05 in. H_2O	Must be leveled carefully; needs calibration
3	Manometer	Medium press diff	0–100 in. H_2O or Hg	0.05 in.	Where used with liquid must be compensated for liquid density
4	Swinging-vane type gage	Moderately low press diff	0–$\frac{1}{2}$ in. 0–20 in. H_2O	5%	Generally usable to atmospheric pressure only
5	Bourdon-tube type	Medium to high press diff, usually to atmosphere	Any	As high as 0.01 psi or 0.01% of full scale	Must be calibrated; subject to damage due to over press-shock or pulsation

Table 5. Measurement of Velocity

No.	Measurement Means	Application	Range fpm	Order of accuracy	Limitations
1	Hot-wire anemometer	Low air velocities; directional and non-directional available	5–1,000	5–20%	Accuracy of some types not good
2	Kata thermometer	Low air velocities in rooms; non-directional	5–300	5–15%	Awkward to use; affected by radiation
3	Smoke puff or air-borne solid	Low air velocities in rooms; highly directional	5–50	10–20%	Awkward to use but valuable in tracing air movement
4	Swinging-vane type anemometer	Air velocities in rooms, at outlets, etc; directional	30–24,000	5%	Not well suited for duct readings; needs periodic check calibration
5	Venturi-type multiplying pitot tube	Low air velocities in rooms and ducts; directional	100–2,000 with micromanometer 180–2,000 with draft gages	1–5%	Accuracy falls off at low end of range; requires calibration
6	Revolving-vane type anemometer	Moderate air velocities in ducts and rooms; somewhat directional	100–2,000	5–20%	Extremely subject to error with variations in velocities with space or time; easily damaged; needs periodic calibration
7	Pitot tube	Std instrument for measurement of duct velocities and pressures	180–10,000 with micromanometer 600–10,000 with draft gages 10,000 up with manometer	1–5%	Accuracy falls off at low end of range
8	Impact tube and sidewall or other static tap	Extremely high velocities, small tubes and where air direction may be variable	120–10,000 with micromanometer 600–10,000 with draft gages 10,000 up with manometer	1–5%	Accuracy depends upon constancy of static pressure across stream section

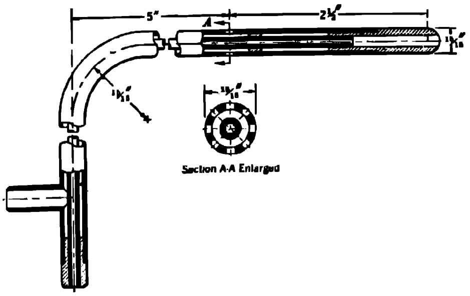

Fig. 13. Standard Design of Pitot Tube

space, some of them are not readily used in ductwork.

For observing velocities in free air in the range of 100 to 1000 fpm, the swinging-vane type of anemometer is probably as satisfactory as any other. The **revolving-vane type** anemometer is also extensively used in this range. However, these instruments are subject to errors as great as 30 to 40%, due to variations in direction and velocity of the air over the face of the instrument. Also, while they are made as a portable instrument, they are very subject to damage in handling. Both types of instruments are sometimes used for measuring velocities in ducts and plenums, but are awkward for this use since the instrument and the observer must be within the duct. Venturi-type multiplying **pitot tubes** are also available for measurement of velocity in this range. These have pressure multiplication factors as high as 10:1.

For relatively high velocities, the pitot tube is considered the standard instrument for velocity measurements. It may be used with a micromanometer, draft gage or "U" manometer, depending upon the velocity and the order of accuracy required. The order of accuracy at the low end of the range is poor but increases rapidly with velocity since the pressure varies as the square of the velocity. A standard form of pitot tube is shown in **Fig. 13.** A pitot tube is directional but is not extremely critical for direction. Where the static pressure of the stream varies with position, care must be used to see that the static holes are at the position for which the pressure and velocity are desired.

For high velocities, for use in very small tubes or where the direction of the air may be variable, velocity readings are frequently made using an **impact tube** together with a sidewall tap for static pressure. The impact tube is customarily a small tube used transversely of the stream. It is closed at the end and with a fine hole drilled close to its end on the upstream side. Static pressures may also be determined by revolving the impact tube through 180°, the true velocity pressures being obtained by multiplying the differential by a factor determined from calibrating tests. Static pressure may also be determined by means of a **Fechheimer tube** having two holes set at an equal and exactly determined angle to the air stream. This instrument, which must be made very precisely, is also subject to variation with velocity.

Volume or Flow Rate Measurement

Means of measuring flow rate are shown in **Table 6.**

Flow of gas and liquids in fair volume through ducts or pipes are most often made by the measurement of the pressure difference across an **orifice, nozzle** or **Venturi tube.** The orifice is more easily changed than are the nozzle or Venturi tube and is less affected by change of Reynolds number. The nozzle is preferred to the orifice by many engineers, due to its relative freedom from influence by approach conditions and the close approach of the nozzle coefficient to 1.0. The Venturi tube is in essence a nozzle with a recovery tail to reduce the pressure drop.

Standards have been established for orifices, nozzles and Venturi tubes covering the shape, surrounding conditions, location of pressure taps and values of discharge coefficient.

Liquid flow may be measured with good accuracy by **timing** a given **weight flow.** While this method is rather generally used in the test laboratory for calibration of other methods, it is particularly useful where the flow rate is low or intermittent and where a high degree of accuracy is required. A convenient direct-indicating flow meter for liquids is the **rotometer,** which comprises a transparent tapered tube in which the rate of flow is indicated by the position assumed by a bob or plug rotated by the liquid.

Table 6. Measurement of Volume or Flow Rate

No.	Measurement Means	Application	Range	Order of accuracy	Limitations
1	Orifice and manometer	Flow through pipes, ducts and plenums—all fluids	Above Reynolds number of 5,000	1%	Coefficient and accuracy influenced by approach conditions
2	Nozzle and manometer	"	"	1%	"
3	Venturi tube and manometer	Same as 1 and 2 above but used where permissible pressure drop is limited	"	1%	"
4	Rotameters	Normally used for liquids	Any	1%	Must be calibrated for the liquid with which used
5	Timing a given weight flow	Liquids only—used for calibrating other flow means	Any	0.1%	—
6	Displacement meter	Relatively small volume flow at high press drop	As high as 1,000 cfm depending on type	0.2–2.0% depending on type	Some types require calibration
7	Gasometer or volume displacement	Short duration tests; used for calibrating other flow means	Total flow limited by available volume of containers	0.5–1.0%	—
8	Element of resistance to flow and manometer	Used for check where there is calibrated resistance element in the system	Lower limit set by readable press drop	1–5%	Secondary reading depends on accuracy of calibration
9	Thomas meter (temp rise of stream due to electrical heating)	Where elaborate setup is justified by need for good accuracy	Any	1%	Uniform velocity, usually used with gases
10	Heat input and temp change with steam or water coil	Check value in heater or cooler tests	Any	1–3%	—
11	Instrument for measuring point velocity	Primarily used in installed systems where no special provision for flow measurements have been made	Lower limit set by accuracy of velocity measurement	2–4%	Accuracy depends upon uniformity of flow and completeness of traverse

For measuring total liquid or gas flow over a period of time, a wide variety of **displacement meters** is available. The Thomas meter has been used in the laboratory for measuring of the flow rate of a large gas quantity with a small pressure drop. The gas is heated by electric heaters and the temperature rise is measured by two resistance thermometer grids. Where the flow of air through heating and cooling coils is being measured, one of the best check methods is through the determination of the heat output or input and the temperature change.

For field measurement of the air flow rate, a frequently used method is to make a **velocity traverse** of the **duct,** using for the purpose a pitot tube and draft gage or other velocity-measuring instrument. Since such an instrument measures the velocity only at a specific point in the stream and since the velocity usually varies from point to point, the accuracy of the reading will depend upon the number of elements into which the area of the duct is divided for determination of the velocity. **Fig. 14** shows typical points for velocity readings in a round and rectangular duct. The round duct readings are taken at the centers of area of equal concentric areas, whereas in the rectangular duct readings are taken in the center of equal square or slightly rectangular areas. Since the lower air velocity around the periphery of the duct is not measured, the air flow reading taken by this method is usually slightly high. In choosing a suitable location for such a traverse, it is desirable to have as long a straight section of duct as possible ahead of the traverse. Where the duct velocity distribution is poor, a greater number of readings should be taken.

Another approximate indication of flow measurement sometimes used for field readings is to take the **pressure drop** across elements having known flow-pressure drop characteristics, such as heating and cooling coils, fans, etc. Such data should normally be used only for check purposes, however, because the resistances of such elements may differ from rating data due to the surrounding conditions.

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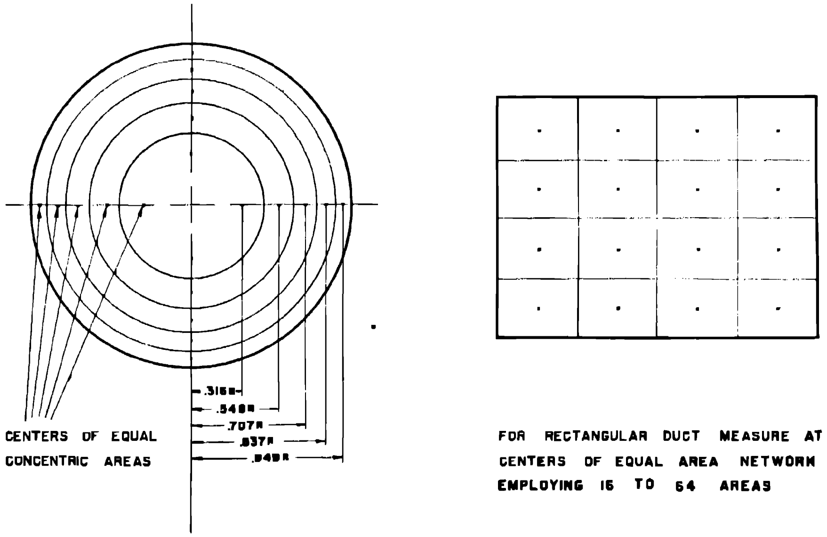
.316R
.548R
.707R
.837R
.949R
CENTERS OF EQUAL CONCENTRIC AREAS
FOR RECTANGULAR DUCT MEASURE AT CENTERS OF EQUAL AREA NETWORK EMPLOYING 16 TO 64 AREAS

Fig. 14. Pitot Tube Traverse Points

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PART IV

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23. COMPRESSORS

Part I. Reciprocating Compressors

THE compressor is one of the five essential parts of the compression refrigeration system, along with the condenser, the expansion valve (or its equivalent), the evaporator and the interconnecting piping. It is the principal moving part, hence knowledge of its construction and behavior is of fundamental importance to the refrigeration engineer.

This chapter discusses the mechanical construction and design of reciprocating compressors. In the selection or design of a compressor, one must analyze the refrigerating cycle as a whole. Once the refrigerant has been chosen, operating temperatures and capacity determined, other features of the compressor follow as a result, including the required displacement, the operating pressures, the materials that can be considered for its construction and the power requirements. Chapter 2 treats of the theoretical cycle and compressor operating characteristics and efficiencies. Combined with this chapter, it gives a comprehensive discussion of the subject.

Classification

1. All reciprocating compressors are essentially alike, consisting of a reciprocating piston and suitable valves, yet their designs vary with numberless possible combinations of details. They may be divided into two main classifications covering the great majority of modern compressors:

a. Single-acting enclosed

b. Double-acting, with crosshead.

The distinctive features of single-acting enclosed compressors are pistons of the trunk type (i.e., driven directly through a pin and connecting rod from the crankshaft without piston rod or crosshead), and a pressure-tight crankcase.

2. Various **combinations** of the following **design features** are used:

a. Number of cylinders—two to as many as 16, sometimes one.

b. Cylinder arrangement—horizontal, vertical, *V*, *W*, or radial.

c. Drive—
   1. Electric motor with rotor mounted on crankshaft inside of crankcase, no moving parts outside.
   2. One end of crankshaft extending out of crankcase sealed with packing or ground joint type of shaft seal and driven by: (a) Electric motor with rotor mounted directly on shaft; (b) electric motor, steam engine or internal combustion engine coupled to shaft; (c) flywheel on shaft belted (V-belt, chain, gear drive) to motor or engine.

d. Lubrication—splash (sizes usually not over 15 hp), or force feed.

e. Suction and discharge valve arrangement—
   1. Uniflow—suction valves in top of hollow piston, with suction gas entering through cylinder walls, discharge valves in head (usual design for ammonia compressors).
   2. Same as (1) but with suction admitted through crankcase.
   3. Suction and discharge valves in head.

f. Type of suction and discharge valves—poppet, ring plate, or flexing.

g. Cylinder cooling—
   1. Water jacket: (a) cylinder wall only; (b) cylinder wall and head.
   2. Air cooled by fins.
   3. No cooling.

h. Head—
   1. Safety head, spring loaded, capable of being lifted off seat by foreign body or liquid.
   2. Bolted tight head.

i. Bearings—
   1. Sleeve.
   2. Anti-friction (usually main bearings only).

j. Capacity control, if provided—manual or automatic.
  1. Variable speed drive.
  2. Adjustable clearance: (a) movable piston in clearance cylinder; (b) one or more fixed-volume clearance pockets.
  3. Bypass from cylinder port (at appropriate fraction of stroke) to suction, preventing compression until covered by advancing piston.
  4. Means to hold suction valve of one or more cylinders open, thus putting them out of service.
  5. Bypass from individual cylinder heads to suction.
  6. Gas bleed from discharge to suction.
  7. Variable stroke.

k. Unloading or starting bypass—
  1. Bypass from discharge to suction with manual or automatic valve.
  2. Suction valves held open.

l. Speed—250 to 3,600 rpm, piston speed usually not over 600 fpm, with 750 as upper limit.

Double-acting compressors are always provided with a piston rod and crosshead. The rod works through a stuffing box in one of the cylinder heads so that the packing is on a reciprocating rod, as compared to the rotating shaft of the enclosed type. Double-acting compressors are built in both horizontal and vertical types. The valves are mounted in the heads or cylinder walls near the head.

In modern designs moving parts are all enclosed to prevent oil splash. Cylinder lubrication and bearing lubrication are separate. Cylinder lubrication may be by injection of oil into the suction gas, feeding oil through the cylinder wall, or a combination of both. Other design features follow those enumerated under single-acting enclosed compressors.

Design Procedure

Before starting the design of a compressor the desired capacity and operating conditions must be known and the refrigerant to be used must be selected. By assuming a reasonable volumetric efficiency, the required swept volume is computed. The type of compressor, whether single or double-acting, high-speed or low-speed, and the number of cylinders, must then be decided upon. It is a matter of simple computation to select the bore, stroke and speed. Average piston speeds are usually kept around 600 fpm.

2. **Calculation** of compressor swept volume is carried out as follows. The nomenclature is this—

h_s = *Enthalpy of vapor at temperature and pressure of compressor suction, Btu per lb
h_e = *Enthalpy of liquid to expansion valve, Btu per lb
M = Capacity in tons refrigeration
v = *Specific volume of vapor at compressor suction, cu ft per lb
V_s = Compressor swept volume, cfm
e_v = Volumetric efficiency, %/100

$$V_s = \frac{200\ Mv}{(h_s - h_e)e_v}$$

3. **Suction and discharge valves.** The limiting factors of speed are the valve area that can be accommodated and the gas velocities permissible without excessive wiredrawing losses. The actual gas velocities vary for different points in the stroke. For design purposes, gas velocity = bore area × average piston speed/valve area. The permissible gas velocity through the restricted area of the valve will depend upon the loss in volumetric efficiency and the excess power requirements the designer is willing to take. In general, with ammonia, velocities up to 12,000 fpm and with Freon-12, velocities up to 6,000 fpm result in no material loss in volumetric efficiency or increase in horsepower. Restricted valve velocities of 20,000 fpm are not unusual. The restricted valve area is not the only factor—the shape of the valve and adjacent ports determines the number of directional changes to the gas flow and affects the efficiency of the conversion of velocity head to pressure head and vice versa, which all affects the pressure drop through the valve.[1]

The ideal valve would meet the following specifications:

---

* The numerical values of the items starred are to be looked up in tables of thermodynamic properties of the refrigerant.

1. Largest possible restricted area, of the shortest possible length
2. Straight gas flow—no directional changes
3. Light weight combined with low lift, for quick action
4. Minimum unbalance
5. No harmful clearance
6. Rugged
7. Inexpensive
8. Tight seating.

Plate valves should have not over 0.075 in. lift. Flexing valves should have their lift restricted, to keep the stresses well below the endurance limit of the steel. Fuchs, Hofmann and Schueler tested plate valves with water and with air, finding that the friction coefficient was materially less with low lifts than with high lifts.[2]

4. The gas **pressure forces** on the piston for various positions, as well as the inertia forces, can be readily computed from **Table 1,** with the following nomenclature:

a = adiabatic factor
A = area of cylinder bore, sq in
A_c = area of cylinder bore, less rod, sq in
d = shaft diam, in.
D = belt wheel diam, ft
E = energy, ft lb
F = total force, lb
F_i = inertia force, lb
F_h = head end force, lb
F_c = crank end force, lb
F_g = net gas force, lb
I = inertia factor
K = tangential factor
k = speed fluctuation, about .02
m = bending moment
N = speed, rpm
P_s = suction pressure, psia
P_d = discharge pressure, psia
R = crank radius, ft
S = stress, psi
T = torque, lb ft
T_e = equivalent torque, lb ft
W_w = wheel rim weight, lb
W = weight of piston, rod, crosshead and connecting rod, lb

The effective connecting rod weight taken into account under W is calculated by multiplying the total rod weight by the distance from the center of gravity to the crank, and dividing by the distance from the crank to the pin. It should be noted that a double-acting piston is doing work on only one face at a time.

For double-acting compressors when

$$P_s a \leqq P_d, \quad F_h = AP_s a \text{ and } F_c = A_c P_s a$$

when

$$P_s a > P_d, \quad F_h = AP_d \text{ and}$$
$$F_c = A_c P_d$$
$$F_g = F_h - F_c \text{ or } F_g = F_c - F_h{}^*$$

For single-acting compressors when

$$P_s a \leqq P_d, \quad F_g = AP_s(a-1)$$

when

$$P_s a > P_d, \quad F_g = A(P_d - P_s)$$
$$F_i = 0.00034\ WN^2RI$$
$$F = F_g + F_i \qquad T = FKR$$

If T is plotted against crank angle in degrees, a curve such as **Fig. 1** results. The mean torque is the area under the curves divided by the length of the diagram.

$$E = \tfrac{1}{2} \text{ area a-b-c-d-e (expressed in lb deg ft)}\pi/180$$

$$W_w = \frac{11{,}744E}{kD^2N^2}$$

For direct-connected motor applications, the crank effort diagram and allowable current pulsation may be submitted to the motor manufacturer.

5 Calculation of **synchronous motor** wr^2 inertia may be carried out with the following nomenclature:

W = Rotor weight, lb
P_r = Synchronizing power, kw per electrical radian
f = Frequency, in cycles per second
C = Compressor factor from **Table 2**
pf = Power factor
hp = Horsepower
e = Motor efficiency

$$\text{Motor input, kva} = \frac{0.746 \text{ hp}}{e \times \text{pf}}$$

$$P_r = 1.7 \times \text{motor input kva (at 100\% pf)}$$
$$= 2.0 \times \text{motor input, kva (at 80\% pf)}$$

$$Wr^2 = \frac{1.34 P_r f C}{(N/100)^4}$$

r = Rotor radius of gyration, ft

6. The maximum **crankshaft** stress must be evaluated treating the shaft as a beam in torsion. For each piston there will be a force F, as determined in Par. 4, acting in line with the stroke. The bending moment on the shaft is determined in both the vertical and horizontal plane from the various forces, the belt pull, wheel weight, etc. The two moments are then combined:

---

* Note that on suction stroke $F_h = AP_s$ and $F_c = A_c P_s$.

$$m = \sqrt{m_1^2 + m_2^2}$$

The torque is then determined at the point where m is a maximum:

$$T_e = m + \sqrt{m^2 + T^2}$$

$$S = \frac{16T_e}{\pi d^3}$$

Sharp corners should be avoided in crankshafts to minimize stress concentrations. The stress should be kept well below the endurance limit of the shaft material. A factor of safety of not less than 3, based on the endurance limit, is suggested.

Since shaft failures are usually due to

Table 1. Constants for Constructing Crank Effort Diagrams

Crank angle	Piston position	Inertia factor I	Tangential factor K	Adiabatic factor, ammonia a	Adiabatic factor, F-12 a
0	.0000	1.200	0	1	1
10	.0091	1.173	.2079	1.012	1.01
20	.0360	1.093	.4065	1.049	1.042
30	.0796	.966	.5870	1.114	1.111
40	.1317	.801	.7421	1.201	1.173
50	.2081	.608	.8657	1.354	1.302
60	.2878	.400	.9540	1.534	1.445
70	.3735	.189	1.0052	1.837	1.696
80	.4622	−.014	1.0199	2.24	2.016
90	.5505	−.200	1.000	2.829	2.469
100	.6358	−.361	.9500	3.718	3.131
110	.7151	−.495	.8743	5.357	4.132
120	.7878	−.600	.7781	7.503	5.765
130	.8509	−.677	.6664	11.87	8.589
140	.9038	−.731	.5435	20.98	14.09
150	.9455	−.766	.4130		
160	.9757	−.786	.2776		
170	.9939	−.797	.1394		
180	1.0000	−.800	.0000		
190	.9939	.797	.1394	1.008	1.007
200	.9757	.786	.2776	1.032	1.028
210	.9455	.766	.4130	1.076	1.066
220	.9038	.731	.5435	1.141	1.121
230	.8509	.677	.6664	1.234	1.200
240	.7878	.600	.7781	1.363	1.309
250	.7151	.495	.8743	1.546	1.461
260	.6358	.361	.9500	1.802	1.668
270	.5505	.200	1.000	2.173	1.963
280	.4622	.014	1.0199	2.727	2.392
290	.3735	−.189	1.0052	3.598	3.043
300	.2878	−.400	.9540	5.049	4.085
310	.2081	−.608	.8657	7.696	5.893
320	.1317	−.801	.7421	13.95	9.882
330	.0796	−.966	.5870	26.84	17.46
340	.0360	−1.093	.4065		
350	.0091	−1.173	.2079		

Crank angles from beginning of stroke towards crankshaft; rod length/crank radius = 5; γ for ammonia = 1.3, γ for Freon-12 = 1.13.

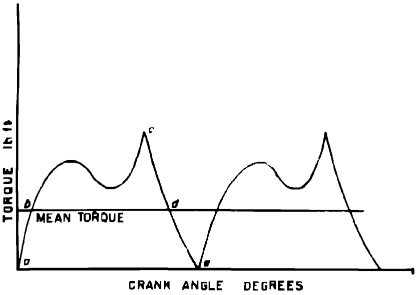

Fig. 1. Crank Effect Diagram for 2-Cylinder, Vertical s.a. Compressor

fatigue, it is suggested to design on the basis of the maximum shear theory. In most compressor designs, assuming the torque as constant at the average value and assuming the maximum bending moment as reversed introduces no significant error. For this condition Mark's Handbook gives the following equation for the shaft diameter:

$$d = \sqrt[3]{\frac{32F_s}{\pi}\sqrt{\left(\frac{Km}{S_e}\right)^2 + \frac{T^2}{Sy^2}}}$$

Where F_s = factor of safety (usually 2 to 3)
K = stress concentration factor in bending which depends upon the relation of the fillet to the height of the shoulder as well as upon the notch sensitivity of the material. With liberal fillets K should not exceed 2.5.
S_e = endurance limit of the material in reversed bending.
S_y = yield stress in tension.

Higher speeds, the elimination of massive foundations, and a more critical public combine to make balancing of crankshafts necessary. Mathematical treatment of this subject can be found in texts on the subject of vibration. (Also see Aldinger and Neubauer, *Refrig. Eng.*, Feb. 1941.)

Shafts have in general been made of forged steel. Some designs call for hardened bearing surfaces to reduce wear. Alloy cast iron such as Meehanite or Gunite is being extensively used for this purpose. It is generally lower in cost than forgings. It has lower notch sensitivity and superior damping qualities. The material has excellent wear resistance in journals but must be highly polished to avoid scratching of the bearings.

7. Bearings. Wristpin bushings are of bronze or cast iron, with pressures up to 1500 psi. The wristpins themselves are case-hardened steel. Crankpin and main bearings are generally white-metal alloys, although anti-friction bearings are also used. On enclosed compressors, thrust bearings must be provided to counteract the force due to crankcase pressure acting

Table 2. Compression Factors for Case in Design[3,4]

Application number	Description	Pulsation, %		
		66	40	20
A.	2-cyl vertical double-acting with 180° cranks for air, Freon, or ammonia. *Full load:* Both cylinders working normally. ¾ *load:* Suction valves lifted on head of one cylinder. ½ *load:* Suction valves lifted on two crank ends, head ends working normally. ¼ *load:* Suction valves lifted on 2 crank ends and one head end. *No load:* All suction valves lifted.	23.0	30.0	
B.	2-cyl vertical double-acting with 180° cranks for air, Freon, or ammonia. *Full load:* Both cylinders working normally. ½ *load:* Suction valves lifted on two crank ends, head ends working normally. *No ¾ or ¼ load.*	24.0		
C.	2-cyl vertical single-acting with ¾-load operation by opening 50% bypass on one cylinder for Freon or ammonia.	14.5	20.0	28.0

on the shaft area at the stuffing box. **Table 3** gives data on white bearing metals. Practice goes considerably below the values given. In enclosed machines, crankpin pressures are generally held below 600 and main bearing pressures below 500 psi of projected area. The trend is away from bearing adjustments and toward precision bearings which require no fitting or scraping. Older practice usually made main bearings 2 to 2½ times the diameter with bearing pressures of about 250 psi. Shorter bearings of length not much greater than diameter will carry higher unit loads.

The parts of the **cylinders and crankcase** are cast of dense high-strength iron, and are carefully tested under hydrostatic and air pressure. Pistons are given a diametral clearance of about 0.001 in. per in. of diameter with modifications for piston material. For high-speed precision-built compressors, the diametrical clearance is further modified for temperature variation of the piston, i.e., greater clearance at the top skirt than the bottom skirt. The crankcase or **frame** supports the shaft bearings and carries the cylinders. It should be checked for stiffness to prevent weaving. Cylinders and crankcase should be hydrostatically tested to two times working pressure and tested with air under water to one and a half times working pressure.

8. Shaft packing. Ammonia compressors are easy to pack, and various soft as well as metallic shaft packings are considered adequate. Freon-12, however, is very difficult to hold, and shaft seals which depend upon a ground seal joint had to be developed. The bellows type is limited to shafts about 2½ in. in diameter and smaller, as it is difficult to secure flexibility of the bellows in large sizes. Some of the other seals can be built for large shafts. The shaft seal, born of necessity for Freon-12 compressors, is now displacing conventional packings on ammonia compressors.

9. Lubrication. Large compressors are

Table 3. Field of Usefulness for Various Bearing Metals[5]

Description of bearing metal	Maximum permissible unit pressure	Minimum permissible Zn/P_{max}	Maximum $P_{max}V$	Oil reservoir temp, F	Minimum crank-shaft hardness	Affected by corrosion?
Tin Base Babbitt Copper 3.50% Antimony 7.50% Tin 89.00% Lead (max) 0.25%	1,000	20	35,000	235	Not important	No
		Standard quality bearings				
Tin Base Babbitt Same composition as above	1,500	15	42,500	235	Not important	No
		Alpha process quality bearings				
High Lead Babbitt Tin 5 to 7% Antimony 9 to 11% Lead 82 to 86% Copper (max) 0.25%	1,800	10	40,000	225	Not important	No
Cadmium-Silver Silver 0.75% Copper 0.50% Cadmium 98.75%	Over 1,800 and up to 3,500	3.75	90,000 and upwards	260	250 Brinell	Not likely if temperature is maintained as specified and proper lubricating oil is used
Copper-Lead Copper 60% Lead 40%	Over 1,800	3.75	90,000 and upwards	260	300 Brinell	

Z = Absolute viscosity of the lubricating oil in centipoises. *Data by Albert B. Willi, Federal Mogul Corporation.*
n = Revolutions per minute.
P_{max} = Maximum load in psi of projected area.
V = Rubbing speed, fps.

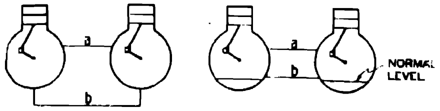

Fig. 2 (Left) and Fig. 3 (Right). Crank Case Oil Equalizing

today all provided with force-feed lubrication; small ones are splash lubricated. The dividing line is usually at about 15 hp. Oil pumps are of the positive displacement type, driven from the crankshaft. Some designs place the pump on the end of the shaft; others submerge it below the oil level. Oil circulating systems are provided with filters.

Enclosed compressors pump some oil. With Freon-12 and methyl chloride the quantity pumped is quite large; it is therefore necessary that it be returned to the crankcase automatically. With a number of compressors in parallel, the returned oil must be properly apportioned between the machines. **Figs. 2 to 5** illustrate various methods of apportioning the oil.

Fig. 2 shows the simplest kind of crankcase oil equalizing system. Line a connects the top of the crankcases to equalize the pressures, and line b connects the bottoms of the crankcases. The oil should have a common level in the two machines. This method is unreliable. A pressure difference of 0.02 psi between the two machines will cause difference in level of about 1 in. Even on the large compressors, a fluctuation of only a few inches is permissible. The very close pressure equalization is difficult to maintain. **Fig. 3** shows a modification of Fig. 2 in that the line b is now at or above the normal oil level. The oil cannot all be forced out of one crankcase through the equalizing connection as only an excess above line b can get across. If, however, the oil is pumped out of the crankcase at the higher pressure, oil cannot

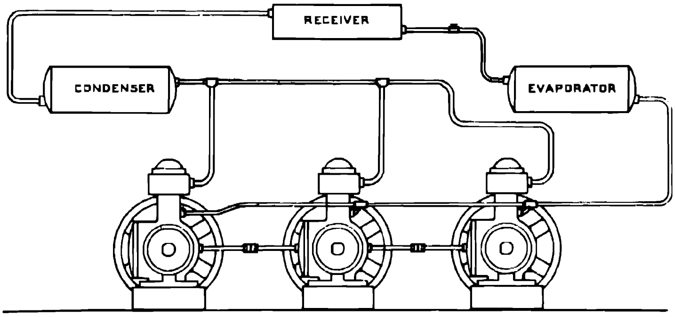

Fig. 4

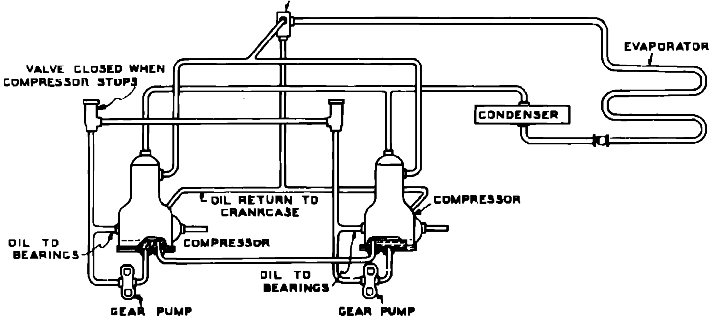

Fig. 5

flow from the other against that pressure, so this system is not absolutely reliable.

Fig. 4 is a modification of Fig. 3 in that a double-seating check valve prevents flow through the oil equalizing line when a material pressure difference exists between machines. Fig. 5 has equalizing connections as in Fig. 3. In addition, the lubricating pump discharges are cross-connected so that if one pump ceases to function due to oil level failure, the pump from the other compressor supplies oil to the compressor whose pump has failed. The electrically operated check valves prevent oil from being pumped into a compressor not operating. **Fig. 6** shows a central reservoir for collecting return oil, and level control means to feed oil to the various crankcases.

Starting

10. Ammonia, Freon-12, and methyl chloride compressors that are to start under load are driven by motors having about 225% starting torque. Ammonia compressors, provided with starting bypasses having not more than 20,000 fpm gas velocity at full speed, are successfully started by synchronous motors having 40% starting and 40% pull-in torque.

Freon-12 compressors when unloaded require from 40 to 60% starting and 50 to 75% pull-in torque. Tests on a two-cylinder single-acting compressor, 7¾-in. bore × 5-in. stroke, equalized at 90 psig at the start with a 2½-in. bypass, required 55 lb ft starting torque, 52 lb ft at 400 rpm, and 60 at 450 rpm, whereas with a 1¼-in. bypass the values were 55, 80, and 97.

11. The **foundation** should extend down to a good solid footing. It may serve to keep various elements, such as bearings, in line, and has the added function, by virtue of its mass, of keeping the amplitude of vibrations small. The heavier the foundation, the smoother will be the operation of the compressor. The vibrational forces are a function of the unbalanced masses and the speed. The size of foundation needed, therefore, depends upon the degree of unbalance. Well-balanced machines can be operated without even being fastened to the floor and with almost no perceptible motion. Only on recently developed high-speed compressors has there been any attempt at balancing. **Fig. 7** gives an average value of concrete foundation volumes for enclosed ammonia compressors ranging from 3 × 3-in. to 12 × 12-in. at speeds of from 500 rpm for the 3 × 3-in. to 300 rpm for the 12 × 12-in. These foundations are for machines without balance weights.

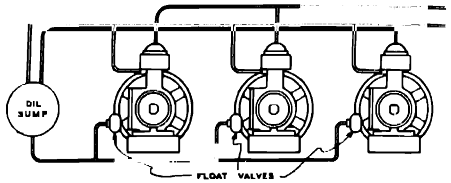

Fig. 6

To avoid transmitting vibrations, foundations should not be tied into building or column footings. To absorb vibrations, machinery may be mounted on cork, rubber, etc., or on springs. For machines having considerable unbalance, better results will be obtained by mounting the machine on a heavy foundation and carrying the foundation on a resilient mounting.

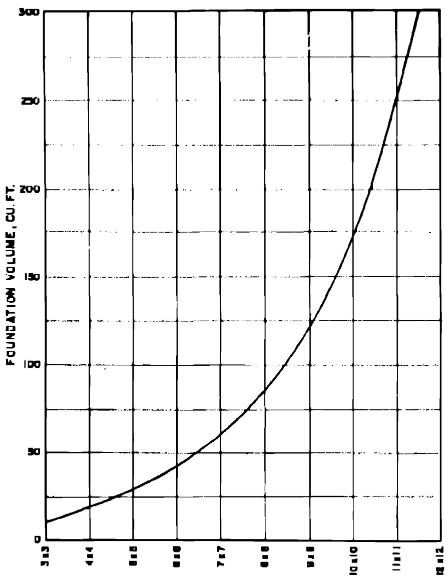

Fig. 7. Foundation Size

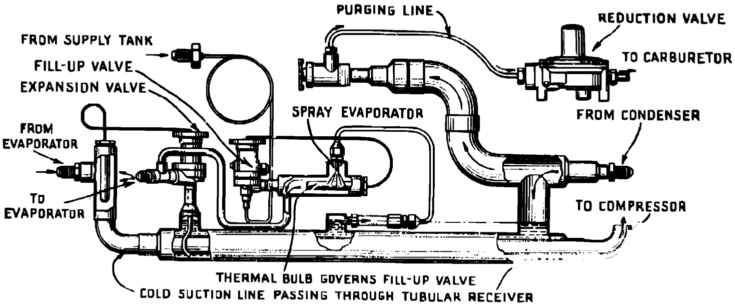

Fig. 8. Propane Unit Equipment

One such mounting can be made by pouring the foundation concrete into a pit lined with corkboard.

Resilient mountings should have a natural frequency much slower than the frequency of the disturbing force (the vibrational forces of the machine). The ratio of disturbing force to natural frequency of the mounting should be at least 1.41, and preferably 4 or more.

Natural **frequency** of the resilient support, in cycles per minute $= 188(1/S_d)^{1/2}$ where S_d is the static deflection, in inches.[9] This represents free motion without damping, such as would result from a spring rubbing in its guide. Damping minimizes the bad effect of operating close to natural frequency and somewhat reduces the isolation efficiency for larger frequency ratios. It may be noted when bringing machines mounted on vibration isolation up to speed, they will shake violently in passing through a certain speed range. That is when the impressed vibrations are close to the natural frequency of the support.

The suction and discharge pressures, or unloading, have no effect on the machine vibration (except the vibration caused by the speed fluctuation of the rotating masses), because the gas pressure forces are balanced out, within the machine. The pressure on the piston imposes a force through the rod to the shaft and bearings into the machine frame. This force is balanced by one of like magnitude on the cylinder head, which is carried back to the frame through the cylinder walls. Unbalanced moving parts are the principal cause of vibration.

The completed design of a compressor is a summation of compromises. Valve area may be sacrificed to keep the clearance within practical limits, the cylinder block design is modified to simplify cleaning, the crankcase is made bigger to give a larger oil reservoir, piping is changed to improve appearance, and weaknesses shown on test must be corrected. The finished result is more than likely the composite product of the best minds of the manufacturer's staff in its effort to provide the best efficiency, reliability and appearance in a compact form, to sell at a reasonable price. If there were one proved best possible compressor others would not be built.

On vertical single-acting machines the clearance between the piston and cylinder head is ordinarily held at about 1/32 in. on all sizes above 4-in. stroke. On a 1¾-in. stroke machine the clearance may be as little as .010 in. Horizontal double-acting machines may be set with a little greater clearance. These values may have to be modified where the design is such as to give unusually great rod expansion under operating temperatures. The clearance should be held as close as practical and yet avoid the possibility of striking the head under any operating condition.

Propane Unit

12. A special application of the refrigeration principle in unison with the use of the refrigerant as a motor fuel has been developed by Schlumbohm, using propane.[*]

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[*] U. S. Patents 1,935,749; 2,082,850; 2,105,387-388-389. There are foreign patents in more than 14 countries.

The cycle is called transitory as the refrigerant passes through the evaporator only on its way to being used in the driving motor, along lines indicated in **Fig. 8.** Propane makes both an excellent fuel and refrigerant.

The carburetor of the engine is connected to the condenser side of the refrigeration circuit. Fueling the engine with condenser gas has the advantage of purging the condenser of non-condensible gases. The condenser is open and connected by the purging line with the carburetor of the propane combustion engine. This purging line is characterized by an ascending tube branching off from the condenser receiver and equipped with baffles to retain oil. There is also a pressure reduction valve, and a fuel regulator, which opens only in response to the vacuum of the engine-intake and which shuts off, if the engine stops.

The low-side is connected with a propane bottle by the fill-up line, which is controlled by two valves: a shut-off valve, which is kept open only if the generator supplies the current and closes if the motor stops, and the "fill-up valve," designed to control the level of liquid in the receiver and to respond to presence of liquid propane at that level. A spray line begins at that chosen level and ends in a secondary evaporator with a very small opening, through which a spray of liquid propane will expand into the secondary evaporator and will produce a cold effect if there is liquid propane at that level. If this is not the case, the spray will run dry and no cold effect will be produced. This cold effect and its cessation are utilized to close or to open respectively the fill-up valve. This valve is a standard expansion valve with the one side of the diaphragm exposed to the evaporator pressure, while the other side is exposed to the pressure of the thermal bulb. The bulb in turn is cooled by the secondary evaporator if there is sufficient liquid propane in the receiver. The fill-up valve is kept closed by this cooling effect. Consequently, lack of liquid propane in the receiver will open the fill-up valve, so that the compressor may take in vapors from the propane bottle until the level in the receiver is filled up by the fresh condensate, which then delivers the spray to produce the cold effect and to close the fill-up valve. A pint of propane is sufficient to circulate in a unit of 1.5 ton capacity.

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Part II. Centrifugal Compressors

1. The centrifugal refrigeration cycle. The centrifugal refrigeration machine, as commercially available in standard sizes, consists of a centrifugal **compressor,** horizontal shell-and-tube **condenser,** horizontal shell-and-tube **evaporator** (water or brine cooler), all interconnected with a high-pressure refrigerant flow **control valve** or other similar means of regulating the flow of refrigerant from the condenser to the evaporator. They are complete unit refrigerating machines arranged for direct drive to either a steam turbine or through speed increasing gears to motor, gas engine or Diesel drives. The refrigeration cycle is the compression cycle with the exception that interstage liquid cooling is incorporated as standard with the machines, and is simply accomplished by flashing liquid refrigerant at condensing temperature to one or more of the intermediate stages, thereby subcooling the liquid before it is supplied to the evaporator. Single-stage cooling is common to most water cooling units, and one- or two-stage cooling may be applied for low-temperature work.

2. The compressor. Centrifugal compressors are similar in many characteristics in design and construction to centrifugal pumps. Erosion or wear does not occur at points of clearance of the impeller and the casing labyrinth as in the centrifugal pumps. No valves, piston rings, or cylinder walls are required as in the reciprocating compressors. Efficiency of the centrifugal compressor is sustained for the life of the equipment. There are no practical pressure limits for the centrifugal compressor, except the limitation of mechanical design and volume flow. The pressure produced depends on the peripheral speed of the wheel, the density of the refrigerant handled, and the number of wheels. It is essential that the centrifugal compressor run at high speeds to compress relatively light fluids such as a gaseous refrigerant as compared with a centrifugal liquid pump. The maximum peripheral speed of the compressor is fixed by the mechanical strength of the materials and the sonic speed found in the refrigerant. Centrifugal compressors are essentially for heavy-duty work, continuous operation and where requirements are for large horsepower. The centrifugal compressor is inherently a constant-pressure machine, designed to give capacity variations and operates with little or no vibration.

Table 4 illustrates an important feature of the centrifugal. Its partial load performance is exceptionally good, which is unusual for machinery in general.

Direct shaft connection is made through a suitable coupling. Speed increasing gears are utilized with electric motors and engines. Constant speed motors, syncronous or induction, may be combined with magnetic or hydraulic slip coupling for variation of compressor speed, either manually or automatically controlled.

3. Refrigerants. Normally, refrigerants in the centrifugal compressor have high molecular weights; however, centrifugal compressors have employed the following refrigerants ("Carrene No. 2" (Freon-11) is a common refrigerant used because of its low pressure and high cycle efficiency):

Ammonia (NH_3)
Dielene (CH_2Cl)
"Carrene No. 1" (CH_2Cl_2)
"Carrene No. 2" (CCl_3F)
Freon-12 (CCl_2F_2)
Freon-113 ($C_2Cl_3F_3$)
Isobutane (C_4H_{10})
Freon-114 ($CClF_2$)
Methyl chloride (CH_3Cl)
Ethylene (C_2H_4)
Butane (C_4H_{10})
Propane (C_3H_{10})
Sulfur dioxide (SO_2)
Water vapor (H_2O)

4. Performance. Centrifugal refrigerating compressors have been built with efficiencies of well over 80%, but most commercial refrigerating compressors of today operate with efficiencies of between 70 and 80%. Centrifugal refrigerating compressors used for water cooling for industrial

Table 4. Performance of Centrifugal Refrigerating Machines Applied to Varying Industrial Loads

Load	%	100	90	80	70	60	50
*Temperature head	%	100	96	92	88	84	81
Speed	%	100	94	90	88	86	84
Bhp	%	100	80	68	59	49	41
Bhp/ton	%	100	89	85	84	82	82
Kw input	%	100	86	76	67	57	49
Kw input/ton	%	100	95	95	95	96	98

* Leaving chilled water and entering condenser water temperatures are constant; gpm of chilled water and condenser water are also constant.

purposes or water cooling for air conditioning applications usually consist of two stages using two single-inlet impeller designs, although some are designed for three-stage operation with three angle-inlet impellers for this type of application. Temperature of chilled water leaving the evaporator is as low as 34 F, and these machines are capable of operating with good efficiencies to high condensing temperatures; thus the differential between evaporating temperature of 25 F (for 34 F water) and 110 F condensing temperature (ordinarily obtained with as high as 95 F entering condenser water) is 85 degrees. This differential of 85 degrees has no particular significance other than to give an indication of the temperature lift at ordinary levels. The ratio of compression, however, for Freon-11 (CCl_3F) between these limits is expressed as follows:

P_2 at 110 F = 28.09 psia
P_1 at 25 F = 4.92 psia

$$\frac{28.09}{4.92} = 5.7$$

$\sqrt{5.7} = 2.40$ per wheel (approx) for two wheels.

For normal water cooling to 45 F (35 F evaporating temp) leaving chilled water temperature, and with 100 F condensing temperature obtained with 85 F entering condenser water the compression ratios are:

P_2 at 100 F = 23.60 psia
P_1 at 35 F = 6.26 psia

$\frac{P_2}{P_1} = \frac{23.6}{6.26} = 3.8$ or $\sqrt{3.8} = 1.95$ (approx) per wheel for two wheels

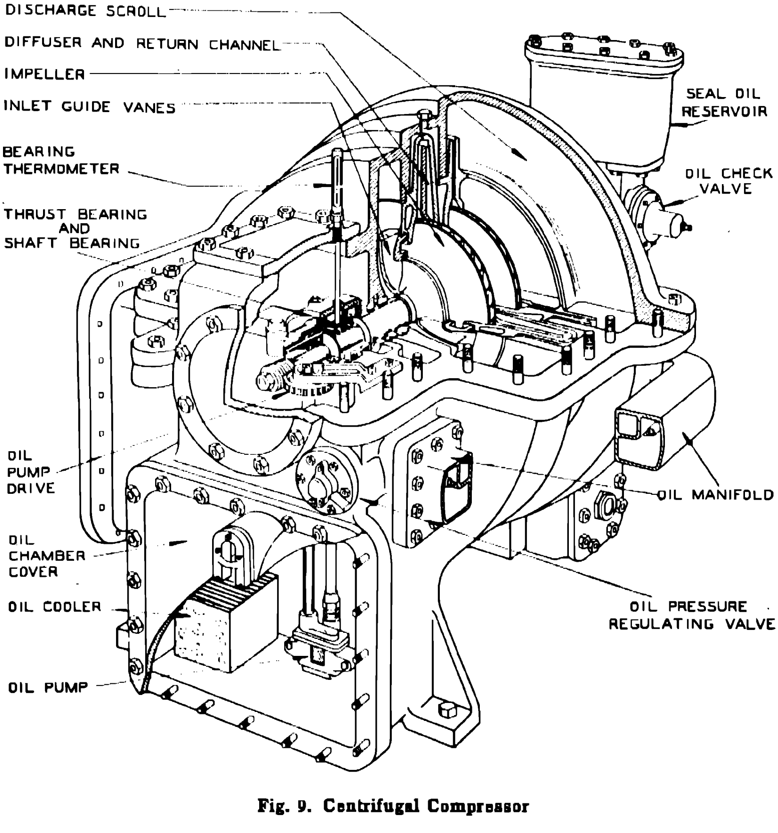

Fig. 9. Centrifugal Compressor

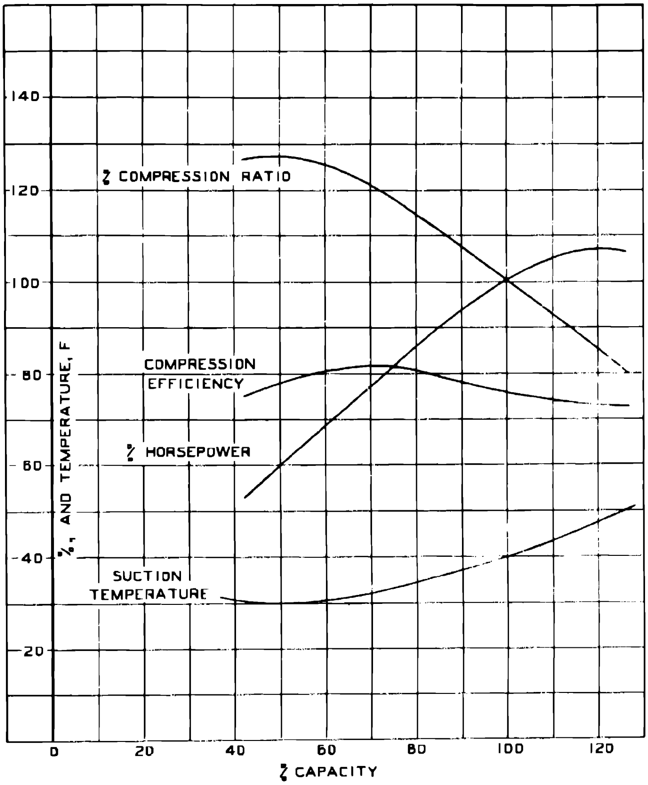

Fig. 10. Performance of Centrifugal Compressor at Constant Speed and Fixed Condensing Temperature

Fig. 10 shows typical performance curves for a centrifugal compressor operating at constant speed and condensing temperature.

Figs. 11 and 12 show performance characteristics of centrifugal machines with different refrigerants and are plotted against varying condensing temperatures. It will be noted that there is an appreciable drop in efficiency and capacity as the condensing temperature increases. This effect can be utilized to govern a machine for reducing capacity, as it is only necessary to raise the condenser pressure; this can easily be done by decreasing the quantity of water flow through the condenser.

Table 7 indicates typical capacities and sizes of the machines of one manufacturer. The capacity in this table is expressed in tons in the three left-hand columns; it will be noted how much the capacity increases in terms of higher suction temperatures.

5. Control. Ordinary safety devices consisting of a **safety relief valve** or valves on the condenser or evaporator are required, along with an **oil-pressure failure switch** for lubrication failure, **high-pressure cutout**, low-temperature water or **brine cutout thermostat**, and sometimes low-liquor **refrigerant temperature** thermostat **cutout.** The operation of the refrigerating compressor to balance with the refrigeration load is not difficult nor critical, and can be accomplished by means of manually varying the quantity of gas entering the impeller of the compressor by suction-damper control or by variable inlet guide vanes. With either motor drive using a wound-rotor motor or with a turbine, variable speed can be accomplished. For the electric

Table 5. Typical Performance of Turbine with a Centrifugal Compressor

Type turbine	Inlet pressure, psig	Superheat, deg F	Exhaust pressure	Steam consumption in lb/bhp-hr
Condensing (high-pressure)	600	300	4 in. Hg abs	8–10
	400	150	4 in. Hg abs	10–12
	250	100	4 in. Hg abs	11–13
Condensing (low-pressure)	1	0	4 in. Hg abs	28–32
Non-condensing	600	300	25 psig	20–22
	400	150	25 psig	27–29
	250	100	25 psig	34–36

drive a sufficient number of steps of resistance to reduce the speed approximately 25%, from full-load to no-load rating, is used. With turbine drive an infinite number of speed ranges can be obtained automatically or manually as desired, by regulating flow of steam by the turbine throttle valve. The centrifugal compressor, like all centrifugal pumping apparatus, can be throttled in capacity simply by increasing the head, which in the refrigeration cycle is accomplished by regulation of condenser water quantity. The regulation of condenser water to increase the head may be accomplished by means of a thermostat located in the brine or in the liquid refrigerant; this was one of the earliest types of capacity control for the centrifugal machine. Although suitable capacity regulation can be obtained by this means, it is more economical from a power and operation cost standpoint to employ speed regu-

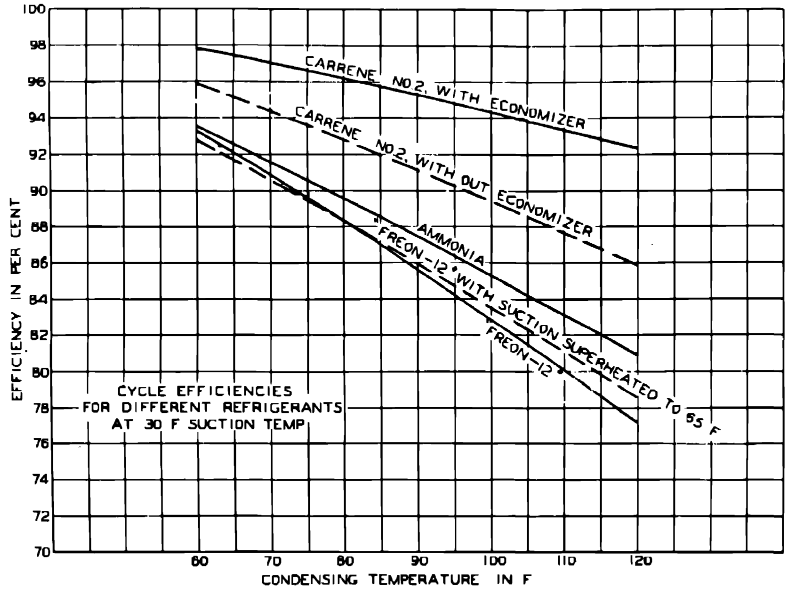

Fig. 11. Cycle Efficiencies for Different Refrigerants under Varying Condensing Temperatures

Table 6. Characteristics of Common Refrigerants

At 5 F sat evaporation and 86 F sat condensing	Gage pressure corresponding to sat evap	Gage pressure corresponding to sat cond	Specific volume of vapor at sat evap	Latent heat of evaporation	Liquid circulation, lb/min ton	Theoretical displacement, cu ft/min ton	Refrigerating effect, Btu/lb	Compression ratio $\frac{\text{Cond pres psia}}{\text{Suc pres psia}}$	Theoretical hp/ton	Coefficient of performance $\frac{200}{\text{Theoretical hp}\times 42.4}$	Efficiency % Carnot cycle
Ammonia	19.6	154.5	8.15	565.0	0.42	3.44	474.4	4.93	.989	4.77	83.2
Freon-12	11.8	93.2	1.49	69.47	3.92	5.82	51.1	4.07	1.007	4.69	81.6
Methyl chloride	6.46	80.00	4.47	180.70	1.33	5.95	150.25	4.48	.963	4.90	85.4
Freon-114	16.14	21.99	4.221	61.98	4.64	19.60	43.10	5.42	1.233	3.83	66.6
Carbon dioxide	317.5	1028.3	0.266	117.5	3.53	0.94	56.7	3.14	1.84	2.56	44.6
Carrene No. 1 (Methylene chloride)	27.55" Vac	8.35" Vac	49.90	162.10	1.49	74.5	134.1	8.59	.965	4.89	85.2
Carrene No. 2 (Freon-11)	23.95" Vac	3.58	12.27	84.00	2.96	36.4	67.5	6.24	.936	5.04	87.8
Propane without economizer	26.98	141.5	2.52	171.35	1.625	4.1	123.	3.75	1.03	4.59	80.0
Butane without economizer	—	26.9	9.98	169.5	1.622	16.2	123.5	5.09	.982	4.81	84.0
Carnot cycle	—	—	—	—	—	—	—	—	.822	5.74	100.0
At 40 F sat evaporation and 100 F sat condensing											
Ammonia	58.6	197.2	3.97	536.2	0.43	1.70	467.8	2.89	.649	7.28	87.3
Freon-12	36.98	116.9	0.792	65.7	3.88	3.70	51.6	2.54	.667	7.07	84.9
Freon-114	0.52	31.69	1.982	58.86	4.47	8.86	44.67	3.04	.878	5.04	64.7
Carrene No. 2 (Freon-11)	15.61" Vac	8.90	5.45	81.2	2.90	15.8	68.8	3.36	.625	7.55	90.9
Water	29.75" Vac	28.07" Vac	2444.0	1068.8	0.198	482.0	1008.9	7.78	.665	7.08	85.0
*Carbon dioxide	553.1	1054.7	0.1444	95.0	5.04	0.727	39.7	1.885	.867	5.45	52.0
Propane without economizer	63.10	174.0	1.33	159.6	1.63	2.17	122.4	2.405	.678	6.98	83.8
Butane without economizer	3.0	37.5	4.88	163.5	1.563	7.64	128.0	2.95	.687	6.89	82.8
Carnot cycle	—	—	—	—	—	—	—	—	.566	8.34	100.0

* For sat evaporation at 40 F and sat condensing at critical point = 87.8 F.

Table 7. Typical Capacity, in Tons, of Centrifugal Machines at Varying Suction Temperatures and Representative Sizes

(Conditions 80 F condensing water, 8 fps water velocity in cooler and condenser)

Tons at			Compressor wheels	Length	Width	Height	Compressor head room	Approximate shipping weight in pounds without drive
35 F	40 F	45 F						
130	145	153	2	14'6"	7'3"	6'4"	6'2"	17,580
145	163	171	2	14'6"	7'3"	6'4"	6'2"	17,900
160	178	188	2	14'6"	7'6"	6'4"	6'2"	18,950
220	245	260	2	14'6"	8'8"	6'9"	7'0"	23,600
246	274	290	2	14'8"	8'8"	6'11"	7'0	24,200
272	300	320	2	14'9	9'2	7'3"	7'0"	25,600
290	324	340	2	14'9"	9'2"	7'8"	7'0"	25,600
376	420	440	2	14'10"	10'9"	8'1"	8'3"	34,400
420	465	495	2	14'10"	10'9"	8'5"	8'3"	35,800
460	515	545	2	15'0"	11'0"	8'9"	8'7"	38,400
500	555	590	2	15'0"	11'0'	8'9"	8'7"	38,400
640	710	750	2	15'8"	12'8"	9'8"	10'6"	55,900
710	790	840	2	15'8"	13'0"	10'0"	10'9"	57,400
780	870	925	2	15'9"	14'0"	10'1"	10'9"	67,000
925	1000	1120	2	15'9"	14'0"	10'1"	10'9"	67,000

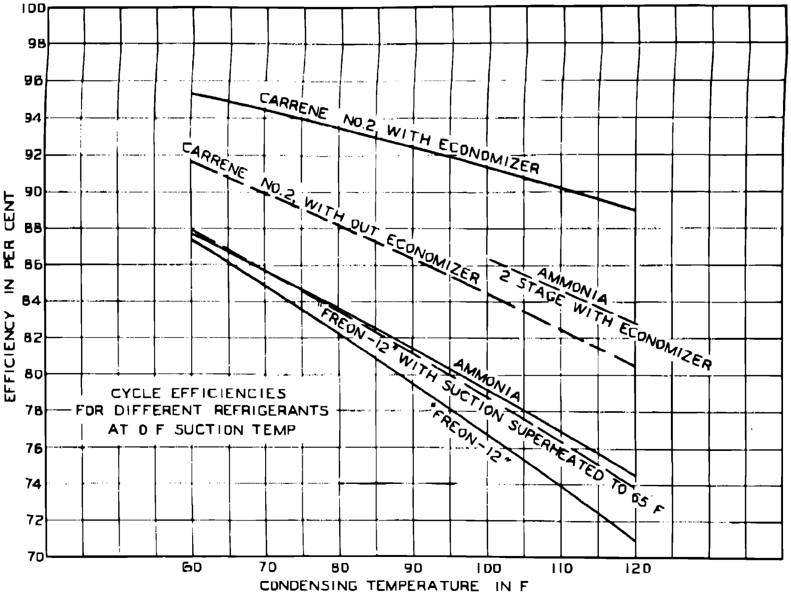

Fig. 12. Cycle Efficiencies for Different Refrigerants Under Varying Condensing Temperatures

lation or regulation of suction gas by means of suction damper control or inlet guide vanes or a combination of both.

Basically a centrifugal machine properly designed does not require extensive control to be capable of **varying with the refrigerating load.** This is accomplished in the design of the impeller and results in a flat characteristic curve for a wide variation in capacity. With this flat characteristic a small change in speed over the entire range of capacity, say from 25% to more than 100%, will have little effect in changing the ratio of compression, and therefore only minor modifications in speed or by suction damper control are necessary for most economical and stable operation.

As the capacity loading of a centrifugal compressor is reduced, a point is finally reached, usually somewhere below 50% of basic selection, where surging occurs. This is a natural characteristic of all centrifugal compressors whether compressing refrigerant vapors, gases or air. The surge point, which may be instantly noted by a change in noise level or hum of the machine, is caused by the compressor removing refrigerant vapor from the evaporator at a rate in excess of its formation in the evaporator. The flow of gas momentarily stops and reverses as the compressor reaches the limit point, and this results in a change of noise level. No appreciable loss of economy and no untoward effects on the mechanical function of the machine are caused by this surging. The point of surging can be lowered by suction damper control, and in severe cases of a low-load condition, usually less than 15%, a bypass can be installed to eliminate surging, but only with some loss in efficiency, and with somewhat higher horsepower per ton required at the reduced load.

6. Ammonia condensing cycle. The complete centrifugal refrigerating unit, consisting of compressor, condenser, evaporator and drive, has been successfully applied to industrial refrigeration not only for cooling of standard brines, such as calcium chloride and sodium chloride, but for all refrigerated solutions. Within the last 15 years, the application of the centrifugal

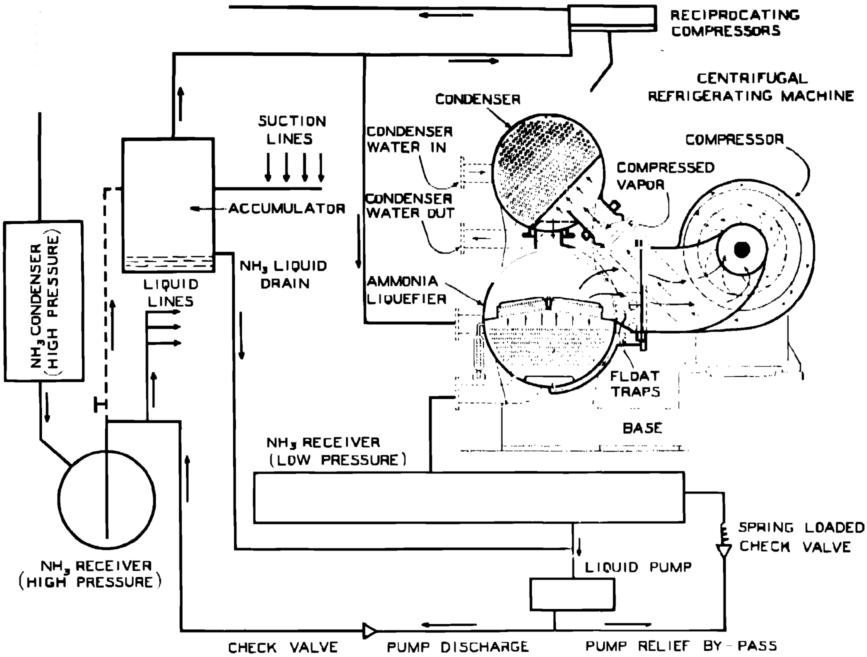

Fig. 13. Ammonia Condensing Cycle Using Centrifugal and Reciprocating Compressors

unit has been broadened to include the condensing cycle for the condensing of various vapors such as chlorine, hydrocarbon gases, but more especially ammonia (see **Fig. 13**).

In 1934 the first application of the centrifugal unit for ammonia condensing was made with success, a design which allowed the cross connecting of a centrifugal refrigerating machine with standard ammonia compression cycles. The merits of the centrifugal machine were thus realized in those plants using ammonia as the refrigerant and having extensive ammonia piping systems.

The ammonia suction line is connected to the evaporator or cooler of the centrifugal unit in a manner identical to that used in connecting it to a reciprocating machine. The ammonia vapor passes through the evaporator tubes, and condenses to liquid by means of heat transfer rather than by compression and heat transfer, as is the case in the standard ammonia condenser. The liquid is taken off at the several passes in the cooler and drained into a conventional receiver, the only difference being that the ammonia is at low pressure and temperature as established by the refrigerating plant requirements. It is obvious that some differential in temperature between the refrigerant evaporating in the cooler and in the liquid ammonia must exist, and coolers are designed for approximately 7.5° to 10° differential for this purpose.

The advantage of this type of ammonia refrigerating system is the elimination of flash gas in the ammonia cycle, with subsequent increase of efficiency of ammonia pipe coils when compared to dry expansion or unit coolers or other evaporators in the plant itself. The lubricating oil ordinarily circulated through the evaporators in a reciprocating cycle is completely eliminated in this type of system, and not of least importance is the safety feature of having the ammonia at low pressure and circulated through the system by means of a conventional centrifugal pump. Ammonia circulating pumps require a 5 to 15-hp motor for supplying approximately 200 to 500 tons refrigerating capacity of ammonia liquid.

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24. CONDENSERS

THE condenser in a refrigerating system converts to liquid form the gas delivered by the compressor, thereby removing the heat of compression and the latent heat of vaporization to provide for a continuous supply of liquid refrigerant for expansion in the evaporator.

The **common forms of condensers** used at the present time may be classified as follows: shell-and-tube, double-pipe, atmospheric and the evaporative type. Principal attention in this chapter is given to condensers as applied in industrial systems. Domestic and commercial-type condensers are also mentioned briefly. In larger installations the shell-and-tube design, built in either vertical open type or horizontal closed type, is more prominent. The horizontal type is exclusively multipass with closed water heads. In the vertical type water flow is by gravity with individual distributors for each vertical tube, thus assuring proper water distribution to each tube.

The type of condenser varies with the size of installation and operating conditions. Shell-and-tube condensers have been replacing the double-pipe or atmospheric type for industrial applications. Evaporative condensers (Chapter 25) have a wide range of application, where water is scarce or expensive. For commercial refrigeration applications, the shell-and-tube, shell-and-coil and double-pipe types are generally used. Domestic and small commercial refrigerating machines use air-cooled condensers which are generally limited to 3 hp condensing unit sizes.

Heat Removed

1. The heat in Btu per min removed in the condenser for each ton of refrigeration (200 Btu per min) produced in the evaporator, may be estimated for various refrigerants from **Fig. 1** for ammonia, **Fig. 2** for Freon-12 and **Fig. 2A** for Freon-22. The values shown are based on saturated vapor at the compressor inlet and adiabatic compression.

The **amount of condensing water** required in pounds per minute may be calculated by dividing the total heat removed per minute in the condenser by the product of the specific heat of water and the temperature rise. To convert pounds of water to gallons, divide by 8.33.

Example: Ammonia condensing temperature = 100 F; suction temperature = 13 F; initial temperature of condensing water, $t_1 = 85$ F; and final temperature, $t_2 = 90$ F. The quantity of condensing water required, gpm per ton of refrigerating capacity, is

$$\frac{H \times 1 \text{ ton}}{8.33(t_2 - t_1)}$$

where $H = 247$ Btu per min taken from curve, Fig. 1; hence

$$\frac{247 \times 1 \text{ ton}}{8.33(90 - 85)} = 5.9 \text{ gpm per ton}$$

Vertical Open Shell-and-Tube Ammonia Condenser

2. The characteristics of this type of condenser are: (a) large condenser capacity may be installed in small floor space; (b) complete installation cost is

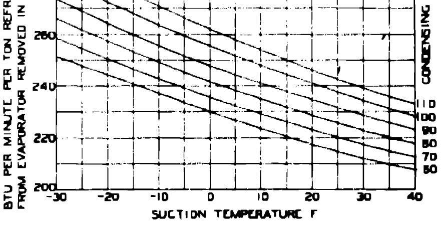

Fig. 1. Btu per Minute Removed in Ammonia Condenser Per Ton of Refrigerating Effect Occurring in Evaporator

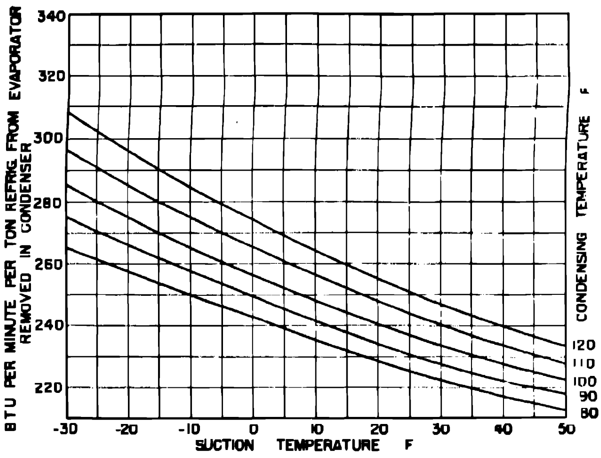

Fig. 2. Btu per Minute Removed in Freon-12 Condenser per Ton of Refrigerating Effect Occurring in Evaporator

low; (c) water distribution is simplified; (d) gas binding difficulties are eliminated; (e) large space is available in the condenser for storing liquid ammonia; (f) it is adaptable to simple purging connections; (g) tubes may be easily cleaned without shutting down; (h) condensed ammonia flows down the tubes and is quickly carried away from the heat-transmitting surfaces while still keeping them wet; (i) condenser can be made to carry large overloads by increasing the quantity of water circulated through the tubes, at the same time avoiding a heavy increase in friction head; (j) condenser is adaptable to any location, inside or outside of buildings, but inside installations have a tendency to "steam" in winter; and (k) it can be used effectively with condensing water from a cooling tower or spray pond.

The laboratory data available on shell-and-tube condensers are by Kratz, Macintire and Gould[2,3] on a 20-in. condenser with 30 2-in. tubes of varied length. Heat transfer coefficients ran from 150 to 350 as the water varied up to 2.5 gpm per tube. Raising the level of the liquid, and thus making the area smaller, increased the unit transfer of heat through the condenser surface.

In computing **transmittance**, values may be based upon logarithmic or arithmetic

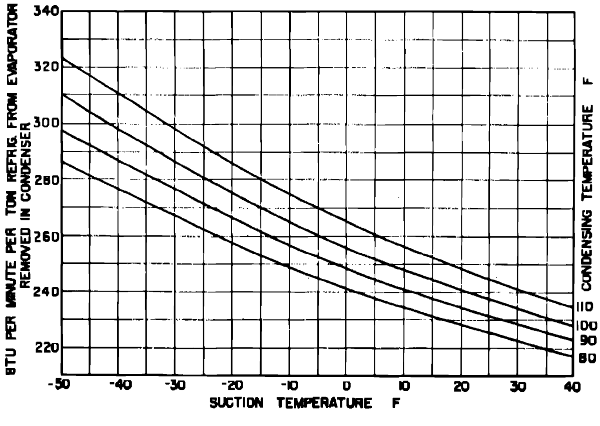

Fig. 2A. Btu per Minute Removed in Freon-22 Condenser per Ton of Refrigerating Effect Occurring in Evaporator

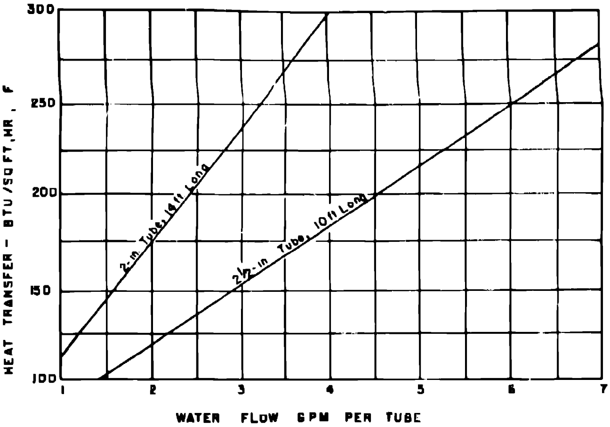

Fig. 3. Heat Transfer Coefficient for Vertical Shell-and-Tube Ammonia Condensers

temperature differences and upon internal or external area. The exact basis used to arrived at a U value is stated here if mentioned in the original source.

In **Fig. 3**, the 2-in. tube curve is presented from the data of Zumbro[4] obtained in tests of a vertical shell-and-tube condenser. Ophuls and Horne[5] gave the results of a certain test made on a vertical shell-and-tube condenser, 10 ft long, having 2½-in. tubes, also in Fig. 3.

Kratz et al call attention to the fact that in their tests, the coefficient of heat transfer at constant water rates was found to decrease with increasing values of the mean temperature difference between the condensing ammonia and the water in the tubes of the shell-and-tube condenser.

The data given in curves and tables by Sloan and Paulener in **Fig. 4** estimate the **performance** of vertical ammonia shell-and-tube condensers. In actual practice they give very close agreement to the data obtained in the field after the condensers have been in service for some length of time, and thus lend themselves to convenient practical use.

The construction of a typical vertical ammonia shell-and-tube condenser is shown in **Fig. 5**. It is built in **sizes** varying from 16 to 60 in. in diameter and in **heights** ranging from 10 to 16 ft, with 14 ft a recommended normal height. As a rule the seamless steel tubes are 2-in. OD No. 11 B.W.G. average wall thickness. The top of each tube is fitted with a water distributor made of cast iron, brass, plastic, glass or porcelain to give the water a swirling motion as it enters the tube.

Example: Required, the surface in a 16-ft vertical ammonia condenser having 139 tons capacity at 10 psi suction and 175 psi condensing pressure, when using 3 gpm per tube at 75 F water temperature. Take 139 tons × 7.26 sq ft per ton × 1.05 × .863 = 916 sq ft of surface. This requires a shell having 109 2-in. OD tubes, 16 ft long, using 327 gpm of condensing water. Tables under Fig. 4 permit correction due to varying tube lengths, gallons per minute water per tube and suction pressures.

Table 1 gives approximately the number of 2-in. tubes which might be placed in various shell diameters.

Numerous **variations** of the **vertical tube** principle have been worked out in ammonia condenser practice. The condenser shell may be made of standard pipe in the smaller sizes and fusion welded shells in the larger. The tube sheets, usually 1

Table 1. Condenser Tubes per Shell*

Shell diam, in.	No. of 2-in. tubes	Shell diam, in.	No. of 2-in. tubes
16	21	38	144
20	33	42	191
24	51	45	220
28	76	48	244
30	85	50	291
34	115	54	330
36	132	60	395

* The number of tubes in each shell varies with individual designs and may vary up to 20% above those shown here.

to 1¼-in. thick, are welded to the shell. One or more grooves are machined in the drilled and reamed tube hole in the tube sheet. Tubes with ground or polished ends are then inserted through their respective tube sheet holes and rolled into the grooves to form a gas-tight joint. This same procedure is followed on horizontal shell-and-tube ammonia condensers.

Fig. 4A shows typical values of leaving terminal difference (temperature corresponding to ammonia pressure minus leaving water temperature) as a function of water flow in gpm per tube for loadings from 5 to 18 sq ft of external tube surface per ton and for condensing water quantities from 1 to 5 gpm per ton. The values are for commercially clean 2-in. OD steel tubes with a nominal length of 14 ft, and are based on a condenser free of noncondensible gases. The rise in the condensing water temperature is considered 28 gal deg per min per ton and the removal of heat from the ammonia vapor as 250 Btu per min per ton.

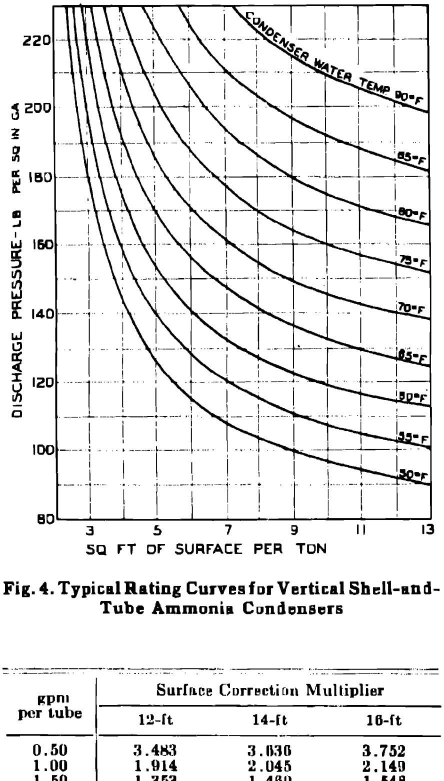

Fig. 4. Typical Rating Curves for Vertical Shell-and-Tube Ammonia Condensers

gpm per tube	Surface Correction Multiplier		
	12-ft	14-ft	16-ft
0.50	3.483	3.636	3.752
1.00	1.914	2.045	2.149
1.50	1.353	1.460	1.548
2.00	1.053	1.142	1.216
2.33	.921	1.000	1.068
2.50	.865	.941	1.005
2.67	.816	.889	.951
3.00	.738	.806	.863
3.50	.657	.720	.774
4.00	.605	.666	.719

Suction pressure	Multiplier	Suction pressure	Multiplier
0	1.12	25	0.98
5	1.08	30	0.97
10	1.05	40	0.95
20	1.00	50	0.93

Condenser selection. As an example, assume that 60 tons of refrigeration are required in the evaporator, and that there are available 240 gpm of 80 F condensing water. The water quantity is then 4 gpm per ton, which fixes the degree rise at 28 gal deg per min per ton, divided by 4 gpm per ton, or 7 F, and the leaving water temperature at 87 F. Fig. 4A shows that 4 gpm per ton at a loading of 10 sq ft per ton would give a leaving terminal difference of approximately 5 F, and that the water flow would be about 2.94 gpm per tube. As the decrease in terminal difference would probably not warrant an increase in surface, giving a condensing pressure corresponding to 93.5 F, or 176.4 psig.

Horizontal Closed Shell-and-Tube Ammonia Condenser

3. The horizontal shell-and-tube ammonia condenser offers the following features: (a) large condenser surface may be installed in comparatively small floor space; (b) total installed cost is low, although not so low as the vertical condenser; (c) it is adaptable to any location in the building or engine room; (d) water piping connections are relatively simple; (e) it must be shut down to remove scale; (f) it can be overloaded but only with a heavy increase in water friction; (g) it can be arranged in batteries for multiple installation, being mounted either side by side or one above the other.

This type of condenser is the standard in many fields of industrial application, especially where location of other types is difficult on account of architectural limitations.

Horizontal condensers are built in sizes

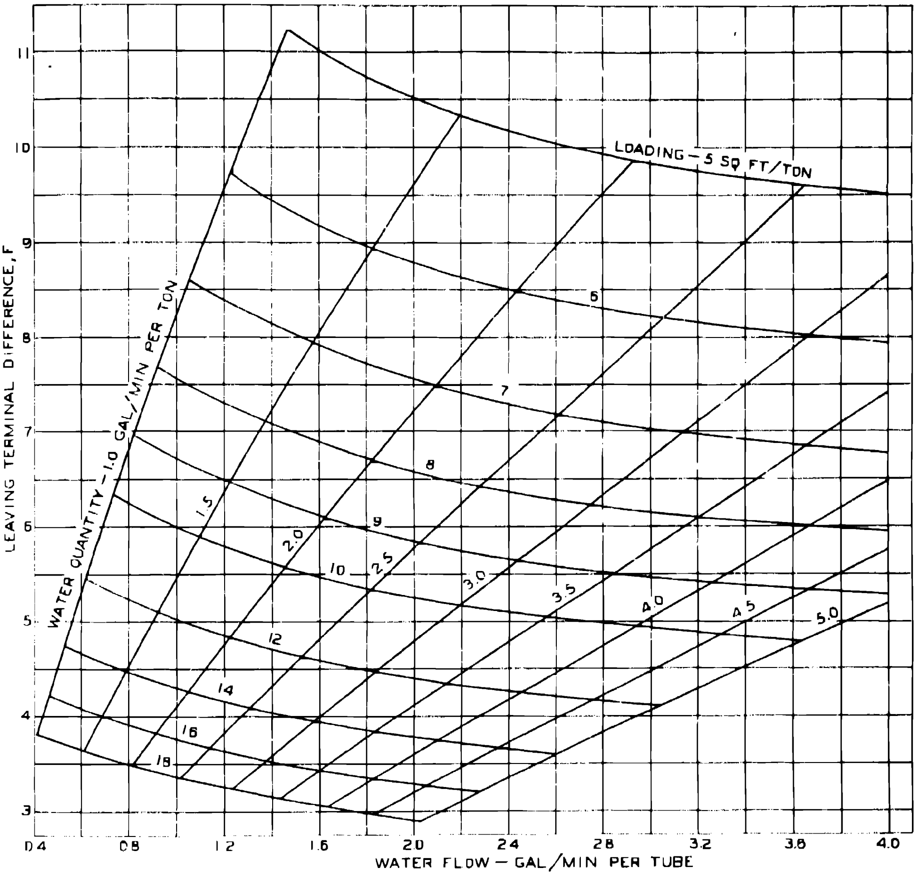

Fig. 4A. Performance of Ammonia Condensers Vertical Open Shell-and-Tube Type for Commercially Clean 2″ OD Steel Tubes, 14′ Long Condenser Free of Non-Condensible Gases[1]

varying from 8⅝ in. to 60 in. in diameter and in lengths varying from 9 to 18 ft in condensers 16 in. and under, and 12 to 18 ft in those larger than 16 in.[1] Seamless steel tubes are usually 1¼-in. OD or 1½-in. OD No. 13 BWG average wall thickness, or 2-in. OD No. 11 BWG average wall thickness.

Figs. 7A and **7B** illustrate a method of rating horizontal ammonia condensers with 1¼-in. and 2-in. OD tubes. The lower two quadrants of Fig. 8A for 1¼-in. OD steel tubes and Fig. 8B for 2-in. OD steel tubes show typical values of leaving terminal difference (temperature corresponding to ammonia pressure minus leaving water temperature) as a function of length of travel (tube length times number of passes), water quantity, and loading (square feet of external tube surface per ton). The curves of the lower left-hand quadrant form a complete picture of heat transfer at a loading of 8 sq ft per ton, the curves of the lower right-hand quadrant offering a correction for other loadings. The values are for commercially-clean tubes in a condenser free of non-condensible gases and are based on a temperature rise in the condensing water of 30 gal deg per min per ton and on a removal of heat from the ammonia vapor of 250 Btu per min per ton.[1]

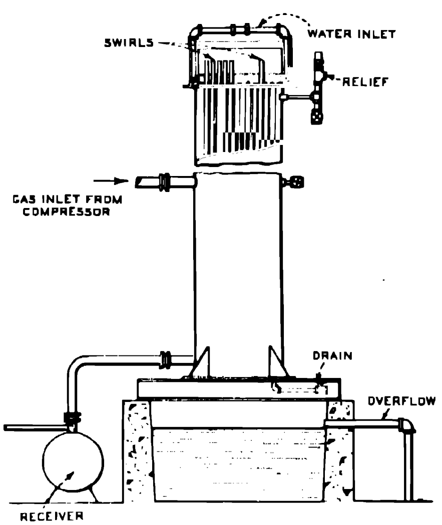

Fig. 5. Construction of Typical Shell-and-Tube Condenser

The condensing temperature is the sum of the entering water temperature, the degree rise, and the leaving terminal difference. It is, therefore, determined by the selection of a water quantity, length of travel, and loading that will give a relatively low condensing temperature at a reasonable pressure drop. As the terminal differences in the accompanying curves are based on a condenser free of non-condensible gases, it is recommended that, for the usual installation, an allowance of 1.5 F be added to the leaving terminal difference for non-condensibles. In refrigerating systems operating below atmospheric pressure, or where unusually high percentages of non-condensible gases are present, this allowance should be increased accordingly.

The upper two quadrants show values of pressure drop as a function of length of travel, water quantity, and loading. The right-hand quadrant need be used only for values of loading other than 8 sq ft per ton. Values in Fig. 7A are based on 1¼-in. OD steel tubes with an average wall thickness of 13 BWG and a nominal tube length of 14 ft; and Fig. 7B on 2-in. OD steel tubes with an average wall thickness of 11 BWG and a nominal tube length of 16 ft. Pressure loss in the water heads is based on Fig. C-5 of the 1939 edition of the Condenser Section, Standards of Heat Exchange Institute, and can be adjusted for other tube lengths if desired.

As an example, assume that 75 tons of refrigeration are required in the evaporator, that the entering water temperature is 80 F, and that the water costs are quite reasonable and the power costs average. For 2-in. tubes (see Fig. 7B), with 5 gpm per ton, the leaving terminal difference (lower left-hand quadrant) is reasonable at a length of travel of 128 ft. For a 128-ft length of travel, the terminal difference would be 5.3 F at 8 sq ft per ton and 4.5 F (lower right-hand quadrant) for 10 sq ft per ton. The resulting pressure drop for a 128-ft length and 5 gpm per ton would be 10.9 psi at 8 sq ft per ton (upper left-hand quadrant) and 7.5 psi at 10 sq ft per ton (upper right-hand quadrant) for a 16-ft condenser. Selecting the lower pressure drop, the condenser would require about 750 sq ft of external tube surface and would have a leaving terminal difference of approximately 4.5 F plus 1.5 F for non-condensible gases, or 6.0 F. The condensing temperature would equal the sum of the entering water temperature, the degree rise (30 gal deg per min per ton divided by gpm per ton), and the leaving terminal difference, or 80 F plus 6 F plus 6 F, or 92 F. This corresponds to a condensing pressure of 172 psig.

Although the optimum selection depends upon the conditions of each installation, the prevailing power and water rates, and, frequently, the quantity of water available, Fig. 7B shows that the above selection

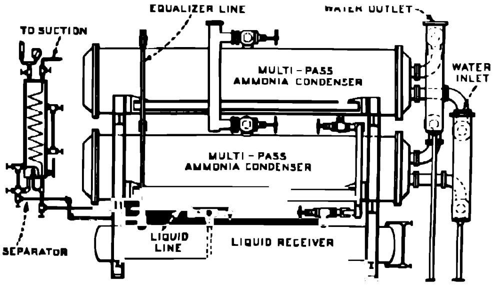

Fig. 6. Horizontal Shell-and-Tube Ammonia Condenser and Receiver

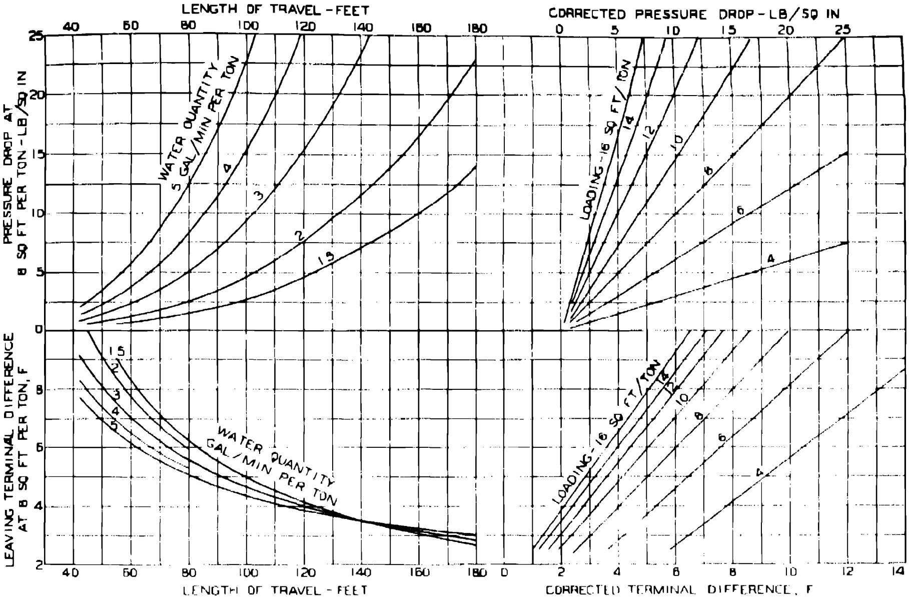

Fig. 7A. Performance of Ammonia Condensers Horizontal Closed Shell-and-Tube Type for Commercially Clean $1\frac{1}{4}''$ OD Steel Tubes Condenser Free of Non-Condensible Gases[1,11]

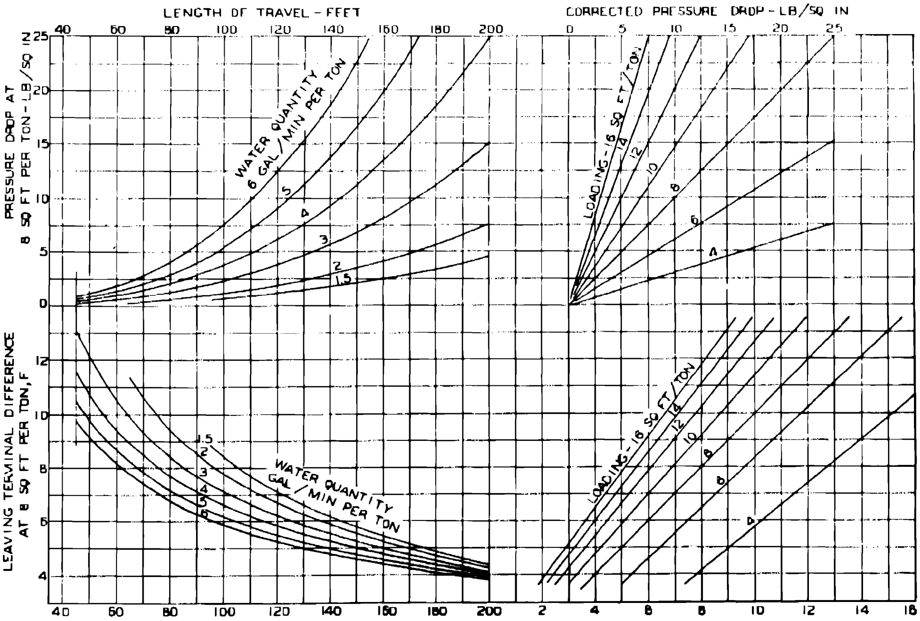

Fig. 7B. Performance of Ammonia Condensers Horizontal Closed Shell-and-Tube Type for Commercially Clean 2" O D Steel Tubes Condenser Free of Non-Condensible Gases[1]

would be a reasonable one for average water and operating conditions. For cases of heavily-incrusted water, the condenser should be selected for low values of terminal difference and on the flatter portion of the curves, where a change in heat transfer will not produce as large a change in condensing pressure.

Figs. 7A and 7B show that at low water quantities, low lengths of travel, and low values of square feet per ton, the leaving terminal difference becomes excessive. At such conditions, a small change in heat transfer would mean a large change in condensing pressure. At the same time, the curves show that, at high water quantities for long lengths of travel and relatively small values of square feet per ton, the pressure drop becomes excessive. The curves therefore afford a direct comparison and a balance of conditions to give a practical and complete analysis of condenser performance.[1]

The most comprehensive data available relating to **heat transfer** of the horizontal multipass shell-and-tube ammonia condenser are given by Kratz, Macintire and Gould.[3] Tests were performed on a bank of two shells, each containing seven 2-in. No. 13 BWG tubes, each 12.82 ft long. The superheat or heat of compression in the gas as it came from the compressor was removed in a gas precooler.

In **Fig. 7** the results have been condensed to give the heat transfer as a function of the water velocity in curve A. This curve refers to the tests made with water circulating parallel through the two shells when the tubes were clean. Curve B refers to data taken in the same manner as those of curve A with tubes which had been in service 16 hr. Curve C refers to dirty tubes; the length of time the condenser had been in service was not stated.

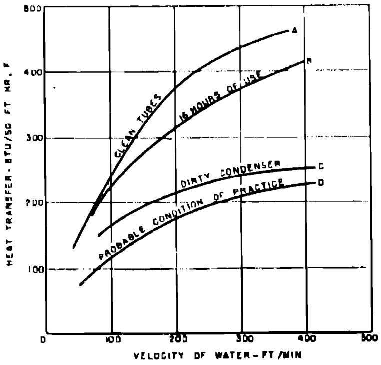

Fig. 8. Heat Transfer Coefficients in Horizontal Shell-and-Tube Ammonia Condensers

The variation in heat transfer as a function of the length of time the condenser has been in service is clearly indicated by the progressively lower values as indicated by curves A, B and C. Ferrous tubes seem to have an increasing scale resistance because of the continuous oxidation of inside tube surfaces, even with normal waters. High rates of hear transfer, often given, are frequently traceable to the fact that such measurements are made on condensers having clean tubes.

Curve D represents the heat transfer that may be expected in practice after the condenser has been in service a reasonable length of time. These curves are based on external tube surface and real temperature differences measured at various points and averaged.[3] The logarithmic mean temperature difference may be used with curve D with sufficient accuracy. This checks with the data of Pownall.[11]

The **water friction** pressure loss, important on horizontal condensers, is composed of the loss through the tubes plus the loss resulting from entry conditions, return bends, and changes in cross sectional area. This is clearly illustrated by **Fig. 9,** taken from the tests of Kratz et al.[3] Note that the loss through the return bends forms a large proportion of the total friction loss, being more than half in this cited case. In order to evaluate this loss, it is necessary to have the return bends resolved into equivalent length of tubing; this value varies principally with design. The values may be estimated in seamless steel tube condensers by using 16 ft of equivalent length of tubing for each return bend, the Williams and Hazen friction factor with an accompanying multiplier of .55 for smooth, clean tubing.

The method of fastening tubes to the tube sheet is the same as on vertical shell-and-tube ammonia condensers.

4. The double-pipe ammonia condenser is usually built of 1¼ and 2-in. pipe from 4 to 12 pipes high in lengths 10 to 20 ft.

Gas comes in at the top, water at the bottom, the action thus being counter-flow. The gas passes between the pipes, the water through the inner pipe. The condenser may be built up with fittings or may be of welded construction.

Numerous references to the heat transfer obtained in double-pipe condensers are available. **Fig. 10** shows data taken from tests by Kratz et al[2] on a standard 1¼-in. by 2-in. pipe, 12 pipes high, 18 ft long condenser. The condenser was new, and hence probably not corroded or scaled. The transfer recorded referred to that part of the condenser surface effective in liquefying the dry saturated ammonia and surfaces ranged from 18.4 to 4.6 sq ft per ton capacity. The results show that the heat transfer varies not only with velocity of the water within the pipes, but with the mean temperature difference between the water and the condensing ammonia. (Note that the water is measured in lb per min.)

Stewart and Holland[3] tested a double-pipe condenser, consisting of 1¼-in. and 2-in. pipes, 8 pipes high, 10 ft long. The heat transfer recorded by these experimenters refers to the overall heat transfer including the removal of superheat. **Table 2** shows the heat transfer through clean surface with a 10 F mean temperature difference for varying water velocities. Actual values should be reduced in accordance with the water fouling effect.

Table 2. Heat Transfer Coefficients in Double-Pipe Condensers

(10 F mean temperature difference)

Water velocity, fpm	Heat transfer, Btu/sq ft hr F
50	185
100	240
150	285
200	330

In these tests the mean temperature difference was obtained by graphical solution of the difference in temperatures between the ammonia and water at various points in the condenser. Both of the foregoing investigators conclude from their tests that in double-pipe condensers the actual heat transfer is a function of both the velocity of the water within the tubes and the mean temperature difference between the condensing ammonia and water, with the heat transfer decreasing with increasing values of the mean temperature difference. **Fig. 11** indicates this variation and may be used to predict double-pipe condenser performance in actual practice.

The **friction head** in a double-pipe condenser is shown in **Table 3.** These are values upon which the pumping equipment may be conservatively estimated.

The large double-pipe ammonia condenser with 2-in. and 1¼-in. pipes is rapidly becoming obsolete in the refrigerating industry; however, special designs still have their economic applications.

5. The **atmospheric** ammonia condenser was used extensively years ago in various forms, such as atmospheric drip type, the straight atmospheric type, with and without drip strips, and the Block type condenser. **Table 4** gives overall values of heat transfer including superheat and con-

Table 3. Water Friction Head in Double-Pipe Condensers, ft

(Pipes 1¼ × 2-in., 20 ft long)

Water, gpm	10-pipe	12-pipe
10	8	9
20	24	27
30	50	59

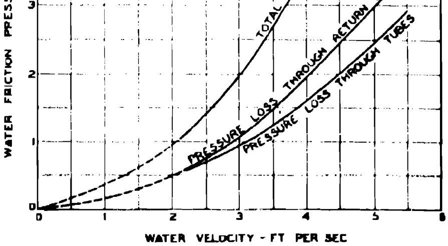

Fig. 9. Water Friction Pressure Loss in Horizontal Ammonia Shell-and-Tube Condensers

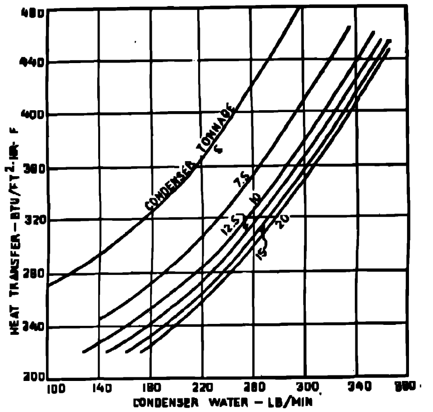

Fig. 10. Performance Curves of a Double-Pipe Ammonia Condenser

forms to those obtained in practice. These values are taken from the data of Torrance and others.

Table 4. Heat Transfer in Atmospheric Bleeder-Type Condenser

Water supply, gpm/ft of length	Transmittance, Btu/sq ft hr F
1	78
2	124
3	153
4	170
5	180

Kratz et al find that in this type the overall heat transfer varies directly with the quantity of water distributed over the condenser, and that the heat transfer is a function of the velocity, independent of the mean temperature difference. The curve shown in **Fig. 12** is taken from their data as giving the percentage of splash loss at different water rates for an atmospheric bleeder condenser provided with drip strips between the pipes. The usual practice in atmospheric condensers is to circulate not more than 2 gpm of water per ft of condenser length; however, with redistributing or splash boards as much as 6 gpm per ft can be used.

For rough calculations of heat transfer, it has been customary to figure 60 to 100 Btu per sq ft hr F as the heat transfer obtained in straight atmospheric type condensers with the gas entering at the top and the liquid removed from the bottom, and 125 for the atmospheric drip, bleeder, or flooded double-pipe type.

The atmospheric type of condenser is today practically obsolete. Several factors have contributed to this end, the foremost being the large floor space necessary for its installation and its higher first cost as compared to the shell-and-tube type condenser. It has, however, the advantage that in winter the water may often be entirely dispensed with and condensation effected by low air temperature. The present evaporative condenser is an outgrowth of this cooling tower and atmospheric condenser combination.

6. Practically all smaller ammonia units use the shell-and-tube condenser. Some of the smaller sizes previously used a shell-and-coil condenser where the water circulates through the coil, but this type of construction causes trouble where bad water conditions are encountered and necessitates a complete condenser replacement, since single tube or coil replacement is impossible. These condensers may be either vertical or horizontal and are sized on the same design data as the larger con-

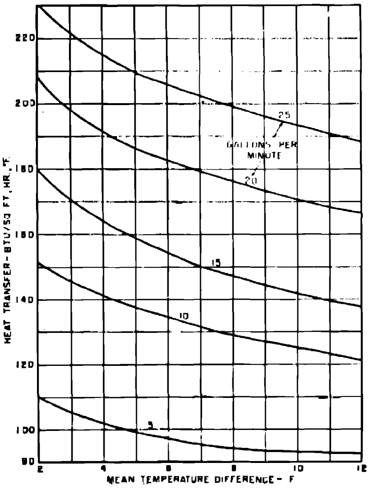

Fig. 11. Heat Transfer Values for Double-Pipe Condensers ($1\frac{1}{2} \times 2$)

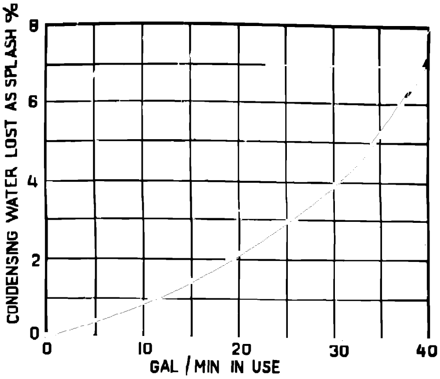

Fig. 12. Water Loss in Atmospheric Condenser, 70F Water, 12 2-in. Pipes High, 20 ft Long

densers. Frequently water velocities are rather low, resulting in lower heat transfer coefficients and much larger amount of surface per ton of refrigeration.

Condensing units operating with the low-pressure refrigerants are sometimes equipped with non-ferrous, double-pipe condensers up to the 2 to 5-hp range; above this shell-and-coil and shell-and-tube condensers are used. (See Chapter 31.)

7. Research on **performance** of ammonia condensers of various types by Linge[7] has brought to light a review and comparison of existing tests on **ammonia condensers.**

Condensation takes place in the superheated part of the condenser as long as its walls are cooled below the condensing temperature. The product of the coefficient of heat transfer and the mean temperature difference between the vapor and the wall is practically the same for superheated vapor and saturated vapor. When the superheat increases, the transmission coefficient decreases to about the same extent as the temperature difference increases; thus the quantity of heat transmission remains about the same. Therefore, an average overall heat transfer coefficient and the mean temperature difference calculated by using condensing temperature corresponding to the saturated condensing pressure and the entering and leaving water temperatures, will give accurate condenser ratings.

It has not been possible to determine with accuracy the coefficients of heat transfer in the **subcooling regions of the different** condensers, because the temperature differences and heat quantities are so small that their measurement is subject to considerable error. These values range from 30 to 60 Btu per sq ft hr F. Hence there is no advantage with big condensers in using a portion of the surface for subcooling which may, however, be advantageously obtained by using a separate cooler.

The main factors affecting the performance of all types of condensers are the **velocity** of the water and the **thickness of the scale** on the tubes. The curves obtained from the observations were always found to be located between those calculated for clean and for scaled condenser surface.

A tabulation of all the test results shows that atmospheric condensers have the slowest water speed and the lowest coefficient of transmission, and that the shell-and-coil condensers with the water through the coil have the highest water speed and the highest coefficient of transmission.

Table 5 from the same source contains factors for conversion of test results obtained with ammonia into values pertaining to other refrigerants. These values were derived by theoretical analysis based on Nusselt's theory of condensation.

Table 5. Relative Heat Transfer with Various Refrigerants under Identical Conditions

NH_3	SO_2	CO_2
100	98	93
200	192	174
400	370	315

Horizontal Closed Shell-and-Tube Freon Condenser

9. Horizontal closed shell-and-tube Freon-12 condensers are built in sizes from about five tons up to those required on the largest installations; however, on these large jobs, greater operating economy is obtained by a multiple condenser installation.

Freon-12 condensers have been made with many types of material specifications, including all prime surface ferrous tubes, and all prime surface or finned, non-ferrous tubes. With the background of performance experience on horizontal ammonia condensers with steel and with the multiplicity of tube types on Freon condensers, efforts were made to resolve the performance of Freon condensers down to their fundamental characteristics.

The computation of **overall heat transfer**[1] in a condenser may be made from test data of heat transfer coefficients of the water and refrigerant sides, physical measurements of the condenser, assumed fouling factors and the following equation:

$$U=\frac{1}{\frac{R}{h_i}+\frac{R}{h_f}+\frac{1}{h_s}+\frac{L}{k}}$$

where

U = overall heat transfer coefficient, Btu per sq ft hr F, based on the external surface and the mean temperature difference between the external and internal surfaces.

R = ratio of external surface to internal surface.

h_i = internal or water side film coefficient, Btu per sq ft hr F

h_f = fouling allowance on water side, Btu per sq ft hr F

h_s = external or refrigerant side coefficient, Btu per sq ft hr F

L = thickness of tube wall, in.

k = thermal conductivity of tube material, Btu in. per sq ft hr F

Note: $\frac{L}{k}$ applies to prime surface. When extended surface is used, this factor should be modified.

Values of h_i are reliably and accurately reported by McAdams.[9] For turbulent flow DG/μ exceeding 10,000 in horizontal tubes and using average water temperatures, he cites the general correlation for condensing:

$$\frac{h_iD}{k}=0.0225\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.4}$$

where:

D = inside tube diameter, ft

k = thermal conductivity, Btu per sq ft hr F

G = weight velocity, lb per hr sq ft of cross section

μ = viscosity, lb per hr ft, based on arithmetic mean temperature of the water = 2.42 × viscosity-centipoises

c_p = specific heat, Btu per lb F

For a given condenser tube and a given range of operating conditions, this equation may be resolved into $h_i = cv^{0.8}$, where c is calculated from the above equation with G resolved into velocity v fps.

Pownall[8] reports that after the oxide film on non-ferrous tubes has been built up with normal waters, its resistance becomes constant and gives a scale factor, h_f, of 4,000, equal to that of non-ferrous tube condensers operating for three years with normal water. Ferrous tubes, however, seem to have an increasing scale resistance because of the continuous oxidation of inside tube surfaces, even with normal water.

For general use, the values of **fouling factors** h_f have been recommended[1] as 2,000 for copper tube and 1,000 for steel tube.

The values of external or refrigerant-side coefficient, h_s, must be determined from test data, since they vary with the physical construction of the tube. Jones[10] cites tests with Freon-11 and $\frac{5}{8}$-in. OD (0.545-in. ID) copper tubes, having sixteen $\frac{1}{16}$-in. high fins per inch, in which the refrigerant film varied markedly and directly with the rate of loading of the surface. The same tests indicated practically no evidence of important changes in refrigerant film coefficient with respect to mean temperature differences or water velocities. These tests were run with high water velocities, 8.1 to 9.2 fps, but because of the improved heat transfer, two passes of water flow are sufficient, where four are usually necessary for plain tubes. Thus the total water friction loss remains about the same. **Fig. 13** is a typical rating. See Paragraph 3 for a general description of the method used for this type of rating curve.

Non-Condensible Gases

10. Non-condensible gases are a combination or mixture of nitrogen, hydrogen, oxygen, chlorine, oil vapors, water vapor, etc., inside the refrigerating system. These gases circulate with the refrigerant, but unlike the refrigerant, they do not liquefy in the condenser, where they collect and on certain types of systems settle in the receiver (See also Chap. 34).

When these gases are present, they will raise the condensing pressure above that corresponding to the condensing temperature by forming a film over the condensing surface or some part of it, thus acting as a resistance to efficient heat transfer.

Due to the continuous flow of the gases entering the condenser, the non-condensible gas film formed around the condensing surface is wiped away and these gases then accumulate in the quiet parts of the condenser, tending to blanket the condensing surface.

This reduction of the effectiveness of the condenser not only causes higher head pressures, but also increases brake horsepower, reduces capacity and promotes oxidation of the oil, especially at the valves due to the presence of oxygen at this point of high discharge temperatures. Operating experience with all refrigerating systems substantiates this. According to Badger,[11] who presents data for steam and air, the film resistance coefficient between steam and a metal wall is as much as six times as great with 4.5% air as with no air present.

As reported by Linge,[7] the presence of these gases according to Dalton's law of partial pressures raises the condensing pressure P by the amount of the partial pressure, P'. The increase of the work of compression is in relation to the ratio $P+P'/P$. This is greater with higher partial pressures and lower condensing pressures.

Non-condensible gases come (a) from outside the system by being drawn into the crankcase of the compressor along with the oil and through leaks in piping, stuffing boxes, etc., when the system is being operated under a vacuum and (b) from inside the system by improperly evacuating a new system before charging, by decomposition of the lubricant under the heat of compression, and by miscellaneous chemical reactions within the system.

A test for non-condensible gases is to shut down the refrigeration system, but to allow the condenser water to flow for a sufficient length of time so that the refrigerant is at the same temperature as the water. If the condenser pressure is higher than that pressure corresponding to the refrigerant temperature, there are non-condensible gases present. Experimental data have shown that with these gases present in the condenser under the above conditions with a 10 F difference between refrigerant temperature and the temperature corresponding to the condensing pressure, there will be as much as 20 to 25 F increase in condensing temperature during operation.

Non-condensible gases can be eliminated to a great degree by purging. This may be

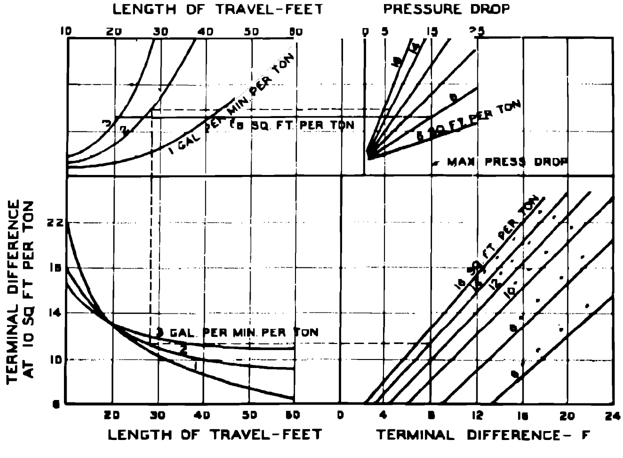

Fig. 13. Rating Curve for Horizontal Freon-12 Condenser with $\frac{5}{8}$-in. OD Tubes Having 8 Fins per Inch, $\frac{1}{4}$-in. High[1, 11]

Example: For 8F difference, 14 sq ft per ton and 2 gpm per ton, find length of water travel and pressure drop. Read vertically on 8° temperature difference line to intersection with 14 sq ft per ton, then horizontally to 2 gpm per ton line. Read 29-ft travel to intersection with 2 gpm and then horizontally to 14 sq ft per ton, finding 8.5 psi pressure loss.

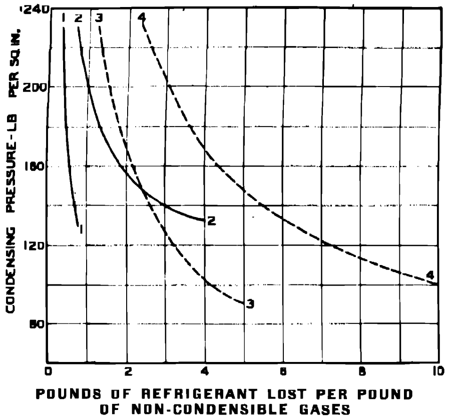

Fig. 14. Ammonia and Freon-12 Loss When Purging at Different Gas Temperatures and Pressures

1—Ammonia, 45F; 2—ammonia, 70F; 3—Freon-12, 45F; 4—Freon-12, 70F

accomplished by shutting down the system as mentioned above, and by blowing off directly into the atmosphere or by using devices for this purpose which work while the system is in operation. Purging connections should be located in the system where non-condensible gases will accumulate. It has been found on long horizontal condensers with the discharge gas inlet in the center of the shell, that these gases will accumulate at the extreme ends and will not work their way down into the receiver. The purging connections should therefore be on the top two extreme ends of this type of condenser.

On any coil type condenser, such as evaporative condensers, where the actual refrigerant gas velocity through the coil is high, the non-condensible gases are pushed down into the receiver and will remain there during normal operation. This type of condenser can be purged very readily from the top of the receiver. However, on ammonia condensers using a number of refrigerant passes in parallel, these gases have been known to pocket in one or more of the passes, rendering them ineffective.

Purging devices employ a refrigerated surface within a container into which is introduced a mixture of refrigerant gas and non-condensible gases. The low temperature surface condenses a great amount of the refrigerant; the remaining mixture is allowed to escape to the atmosphere. **Fig. 14** shows the refrigerant loss when purging ammonia and Freon-12 at various temperatures and pressures, and indicates the desirability of purging devices. Even though Freon-12 losses are considerably greater than ammonia, it is economical to purge these systems as thoroughly as ammonia systems. This can be determined by balancing the savings in compressor power requirements and increased wear on machinery against the loss of refrigerant.

Although it is desirable to operate without non-condensible gases, exceedingly low condensing pressures are also to be avoided. In the endeavor to reduce condensing pressures, the minimum pressure must be taken into account; i.e., the pressure below which the system will not operate satisfactorily because of existing static heads and resistances of mains and control apparatus to the flow of the refrigerant. Condensing pressures are best increased for this purpose by decreasing the water flow.

Fouling Factors

11. Frequently the progressive decrease in capacity of shell-and-tube or shell-and-coil condensers is due to the fouling on the water side of the tube. In order that condensers will have sufficient excess tube surface to maintain satisfactory performance in normal operation, with reasonable service time between cleanings, it is recommended that standard minimum fouling factors be used. For conditions of extreme fouling, poor maintenance and the like, higher fouling factors should be used. These factors are the reciprocal of the heat transfer rate of the layer of foreign substances deposited on the surface of the heat transfer tubing, and enter into the equation as explained under Freon condensers. In order to obtain the total resistance to heat transfer, the sum of fouling resistance as selected is added to the resistance of the fluid film on the inside of the tubes and on the outside of the tubes and the resistance of the tube wall. The effect of fouling resistances in reducing clean coefficients is shown in **Fig. 15. Table 6**[12] lists the fouling factors for various types of water.

12. **Design water temperatures** for condensers are tabulated in Chapter 11 along

with the source of the water for most cities in the United States over 20,000 population. These values are the highest design water temperatures occurring from June through September; however, this temperature occurred in August in the majority of the localities. The **design wet bulb temperatures** for selection of evaporative condensers or cooling towers also have been included. It may be desirable to check local conditions for water and wet bulb temperatures.

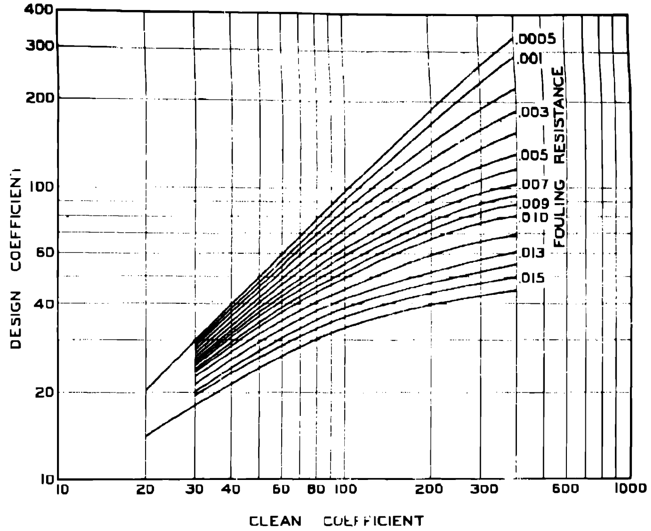

Fig. 15. Effect of Fouling Resistance on Heat Transfer Coefficients

13. Where state or municipal codes do not take precedence **design pressures, materials, welding, tests** and **relief devices** are recommended to be in accordance with ASME Unfired Pressure Vessels Section of the Boiler Construction Code and ASRE Standard 15, which is the American Standards Association Code B9, Safety Code for Mechanical Refrigeration.

Table 6. Fouling Factors for Various Types of Condenser Water

Types of water	Fouling factor
Sea water	.0005
Brackish water	.001
Cooling tower and artificial spray pond:	
Treated make-up	.001
Untreated	.003
City or well water	.001
Great Lakes	.001
River water	
Minimum	.001
Mississippi	.002
Delaware, Schuylkill	.002
East River and N.Y. Bay	.002
Chicago Sanitary Canal	.006
Muddy or silty	.002
Hard (over 15 gr per gal)	.003
Engine jacket	.001
Distilled	.0005

14. The condenser, being a part of the refrigeration cycle high-side, is necessarily used with other more or less essential accessories; of these the important items are the **liquid receivers** and **oil separators.**

In some cases the condenser may also be used as the liquid receiver, but a separate receiver is more frequently advisable and necessary, since liquid accumulation in a condenser blocks off condensing surface with a consequent reduction in capacity or an abnormal increase in condensing pressures. It is desirable to have the liquid receiver of such size that it can hold the entire refrigerant charge, but this is impossible at times, especially on systems with large flooded evaporators. In all cases, the minimum receiver volume should be such that it can absorb the seasonal fluctuations in liquid refrigerant requirements; the liquid seal in the receiver must never be broken, and the liquid level should never build up into the condenser. Although the receiver is a part of the high-side, its size is predicated upon the design and operation of the low-side.

Sizes recommended[1] for ammonia liquid receivers are from 8 in. to 36 in. in diameter, with lengths varying from 6 to 16 ft, depending upon diameter. Freon-12 liquid receivers are built in the same sizes with connections suitable for Freon-12 mains. Many special receivers are made for evaporative condensers, and condensing units with design efforts made to balance the size of the receiver with the capacity of the unit so that the receiver will fit the majority of installations.

Discharge oil separators are traps installed between the compressor and condenser to remove the major portion of the compressor oil carried along by the superheated discharge gas. These separators introduce large volumes into the discharge main to reduce gas velocities, thus permitting the oil particles to be separated by filter, centrifugal, or impingement type separators. The oil removed in the separator prevents fouling of the condenser and evaporator and results in an increased plant efficiency. These separators should be installed close to the condenser so that the pulsating effect of the compressor is reduced as much as possible and also to permit the superheated vaporized oil to form into droplets.

The separated oil on ammonia or Freon-12 installations may be drained automatically back into the compressor crankcase; however, on ammonia plants it is customary to discard this oil or reclaim it by acceptable methods.

Mean Temperature Difference

15. When there is any great change in the temperature difference between two media at different parts of the surface of heat transfer apparatus, it is customary to use the *logarithmic*, rather than the *arithmetic*, **mean temperature difference** in computing heat transfer. This is based on the assumption that the coefficient of heat transfer for all parts of the surface is constant, and that therefore the rate at which heat is transferred through any portion of the surface is proportional to the temperature difference existing there. The formula is

$$D = \frac{\Delta_1 - \Delta_2}{\operatorname{Log}_e \dfrac{\Delta_1}{\Delta_2}}$$

where D = logarithmic mean temperature difference

Δ_1 = difference between condensing temperature and entering water temperature

Δ_2 = difference between condensing temperature and leaving water temperature

Solution requires reference to a table of natural logarithms, which is not always at hand. The formula may also be written

$$D = \frac{\Delta_1 - \Delta_2}{2.3026 \log_{10} \dfrac{\Delta_1}{\Delta_2}}$$

in which case a table of common logarithms is suitable.

Also, it frequently happens that either Δ_1 or Δ_2 is not known, but D is known; or that neither Δ_1 nor Δ_2 is known, but that D and $(\Delta_1 - \Delta_2)$ are known; or that $\left(\dfrac{\Delta_1 - \Delta_2}{D}\right)$ is known, together with Δ_1 or Δ_2. In any of these cases a trial and error solution is required.

Fig. 16 was prepared by A. B. Stickney[13] to eliminate the need of a table of logarithms, to facilitate computation, and to obviate the need of trial and error solutions where possible and otherwise to simplify them. By its use, knowing any two of the five quantities D, Δ_1, Δ_2 $(\Delta_1 - \Delta_2)$, $\left(\dfrac{\Delta_1 - \Delta_2}{D}\right)$, the other three can be read directly.

In refrigerating engineering, the most frequent cases involve either condensing a refrigerant by heating a fluid, or cooling a fluid by evaporating a refrigerant. In these cases the formulae to be solved take the form

$$sUD = wc(t_1 - t_2) = w_r r \text{ in which}$$

s = area of heat transfer surface, sq ft

U = coefficient of heat transfer, $\dfrac{\text{Btu}}{\text{sq ft hr F}}$

D = logarithmic mean temperature difference

w = weight of fluid, lb per hr

c = specific heat of fluid

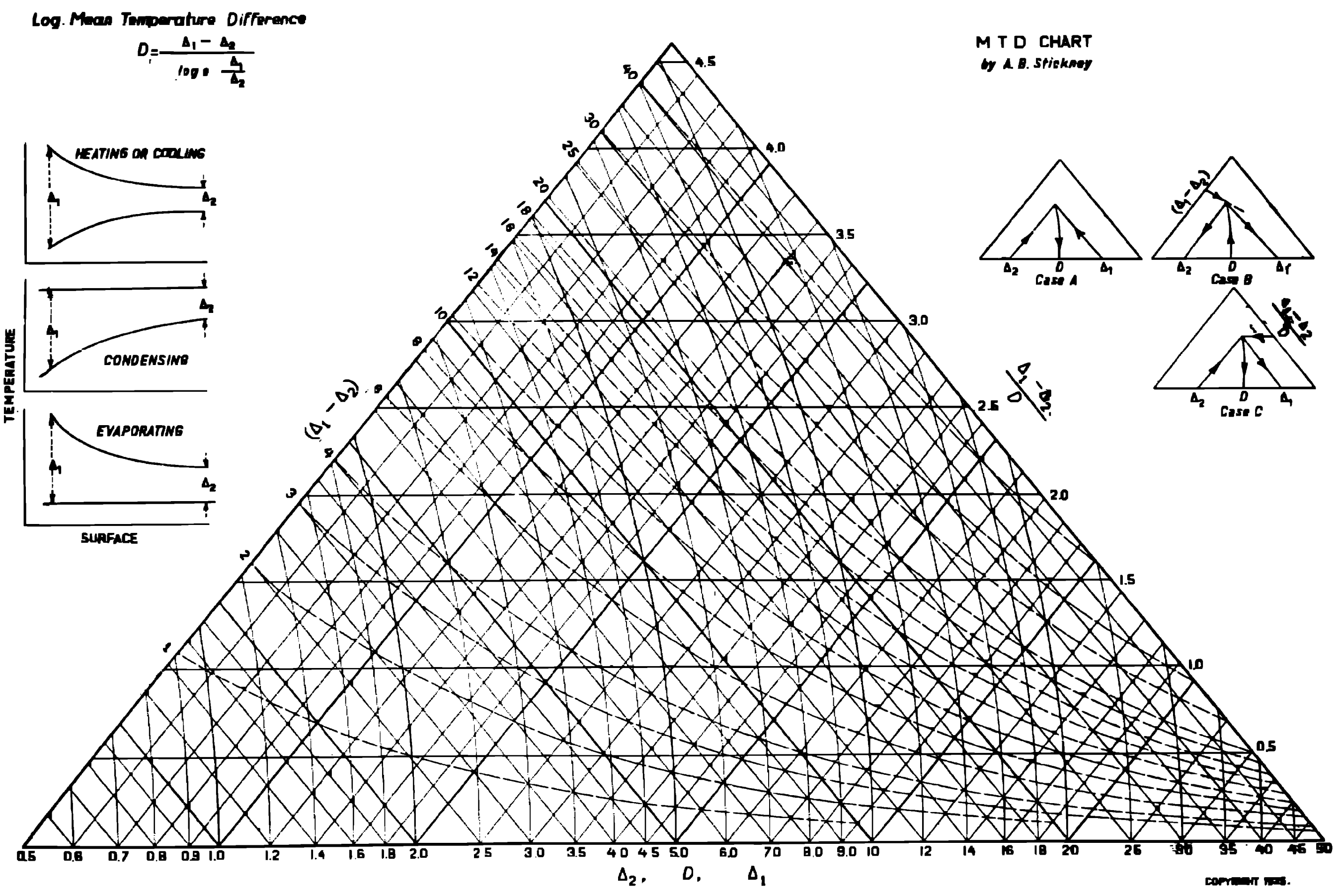

Fig. 16. Chart for Determination of Logarithmic Mean Temperature Differences

t_1 = temperature of fluid being heated leaving apparatus, *or* temperature of fluid being cooled entering apparatus

t_2 = temperature of fluid being heated entering apparatus, *or* temperature of fluid being cooled leaving apparatus

w_r = weight of refrigerant evaporated or condensed, lb per hr

r = the change in heat content in Btu per lb of the refrigerant.

Since the condensation or evaporation takes place at constant temperature,

$$(t_1 - t_2) = (\Delta_1 - \Delta_2)$$

Examples:

A. Suppose a condenser in which the temperature of the "water on" is 70 F, of the "water off" is 80 F, and of the condensing refrigerant is 85 F. Then $\Delta_1 = 85 - 70 = 15$; $\Delta_2 = 85 - 80 = 5$. To find D, read 15 at the bottom of the chart, and follow diagonal upward to the left. Read 5 at the bottom of the chart, and follow diagonal upward to the right. From where these intersect, follow solid curve down to bottom of chart and read $D = 9.1$.

B. A condenser having 1,000 sq ft of surface must handle a load of 100 tons, using 300 gpm of water coming on at 70 F. The coefficient of heat transfer is 150. What is the condensing temperature of the refrigerant? Taking 30 gal deg per ton, the temperature rise of the water is $(100 \times 30)/300 = 10°$ $= (\Delta_1 - \Delta_2)$. 300 gpm is 150,000 lb per hr, the specific heat of water is 1, and therefore

$$D = \frac{150{,}000 \times 1 \times 10}{1{,}000 \times 150} = 10 \text{ F mtd}$$

To solve for Δ_1 and Δ_2, read in $(\Delta_1 - \Delta_2) = 10$ at left of chart, and follow dashed curve to right. Read in $D = 10$ at the bottom of chart, and up along solid curve. From where these intersect, read down diagonally to left and right to find, respectively, $\Delta_2 = 5.8$ F and $\Delta_1 = 15.8$ F. The temperature of the condensing refrigerant is $70 + 15.8 = 85.8$ F.

C. Given the same condenser, the same quantity and temperature of water, and the same coefficient of heat transfer, what tonnage will give a terminal temperature difference of 3 F? And what will be the temperature of the condensing refrigerant?

$$\left(\frac{\Delta_1 - \Delta_2}{D}\right) = \frac{sU}{wc} = \frac{1{,}000 \times 150}{150{,}000 \times 1} = 1$$

Read $\left(\frac{\Delta_1 - \Delta_2}{D}\right) = 1$ at right of chart, and read to left along horizontal line. Read $\Delta_2 = 3$ at bottom of chart, and read upward to right. From point of intersection, read downward to right to find $\Delta_1 = 8.2$, and read to left along dashed curve to find $(\Delta_1 - \Delta_2) = 5.2$ F. Taking 30 gal deg per ton, the tonnage is $(300 \times 5.2)/30 = 52$ tons, and the temperature of the condensing refrigerant is $70 + 8.2 = 78.2$ F.

Although the above examples have been confined to condensers, cooler problems with an evaporating refrigerant can be solved by the use of the chart along exactly the same lines. In addition, the chart will be found useful in the solution of problems involving heat exchange between two fluids each changing in temperature, providing two of the five quantities are known. In case they are not and a trial and error solution is necessary, the chart can be used in solving the various trial values.

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25. EVAPORATIVE CONDENSERS

EVAPORATIVE condensers are of comparatively recent origin as compared to water-cooled condensers or cooling towers. The principle of operation is shown in **Figure 1**. Water is sprayed over the outside of the condensing coil, drips into the water tank where a pump returns it to the spray heads. Air passing over the coils cools the water by evaporation. Make-up water to the tank is generally supplied through a float valve to maintain a constant level.

The eliminators shown above the spray nozzles must be completely effective in preventing drift of water particles. Such drift causes corrosion of unprotected metals and tends to deposit scale on the fan blades which may eventually interfere with proper operation. Condensers are also available with the blowers located on the entering side as indicated in **Figure 2**. In this design effective eliminators are also required in order to prevent drift of water particles into the surrounding area. Evaporative condensers are available as factory assembled units in sizes from 1 to 150 tons capacity.

Water Consumption

The total heat discharged by a condenser is always greater than the refrigeration load by the heat of compression. Nominal rating at standard* conditions assumes a ratio of 1.20 or 14,400 Btu per hr of condenser load for each ton of refrigeration load. This ratio of the condenser load to the refrigeration load varies with the condensing and evaporating pressures as indicated in Figures 1, 2 and 2a, Chapter 24.

Almost the entire load is removed in the form of latent heat of the water that is evaporated. Each pound of water vaporized absorbs approximately 1,050 Btu thus requiring 13.7 lb or 1.65 gal of water per hr per ton of refrigeration load at nominal conditions. The actual water consumption will be greater than the theoretical con-

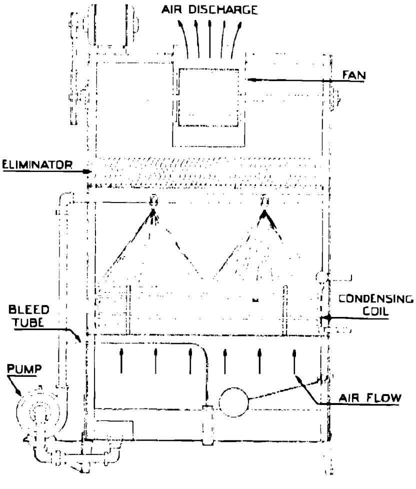

Fig. 1. Functional View of Conventional Evaporative Condenser

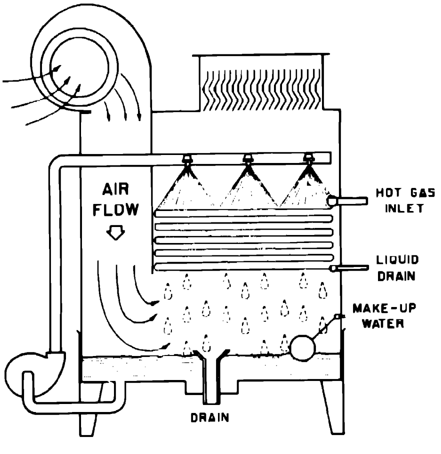

Fig. 2. Functional View of Blow Through Type of Evaporative Condenser

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* See page 442.

sumption due to the necessity of wasting a portion of the water in order to prevent excessive concentration of impurities after continued operation. Heavy scale formation, corrosion and other operating difficulties are likely to result if the concentration of impurities is not kept reasonably low.

On large installations it is possible to remove the impurities by blowing down or draining out all of the water in the sump pan at regular intervals. More generally however some form of automatic bleed is provided whereby a small amount of water is continuously drained away during operation of the pump. In Figure 1 this is accomplished by a small bleed pipe connecting from the pump discharge to the drain. It is necessary to select the size of this bleed line carefully or to use a valve to adjust the flow. In Figure 2 the bleed is accomplished by a funnel whose diameter determines the amount of water diverted to the drain.

The desired amount of water wasted depends upon the concentration of impurities and other chemical properties of the water supply. In good water areas it may be as little as one-half the amount evaporated. With badly contaminated water a much higher rate of waste should be used. Thus for good conditions the total water consumption of the evaporative condenser will be approximately three gal per hr per ton of refrigeration load. It will generally be less than 5% and seldom over 10% of the water consumed by a water-cooled condenser.

The formation of scale deposit may be minimized by chemical treatment of the water. A common method of treatment is to use a slowly soluble vitreous phosphate in a feeder in the water supply or in a wire basket located in the sump pan. The amount of the phosphate granules in the feeder or basket determines the rate of treatment regardless of the amount of water recirculated. Thus the rate of treatment is controlled by replenishing, at intervals, a suitable amount of phosphate in the feeder basket.

In some areas, in addition to the formation of scale evaporative condensers may be subject to corrosion and the accumulation of algae and slime. These conditions can also be minimized by chemical treatment. It is generally desirable to handle such treatment under the supervision of a trained water chemist.

Heat Transfer Processes

Two distinct heat transfer processes occur in the evaporative condenser. (1) From refrigerant to the tube wall, through the tube wall and water film to the water surface, heat transfer depends on the difference between the temperature of the condensing refrigerant and the temperature of the wetted surface. (2) Heat flow from the wetted surface to the air is in the form of both sensible and latent heat (moisture vapor). The combination of these two can be expressed in terms of the difference in the enthalpy of saturated air at the water surface temperature and the enthalpy of the air passing over the tubes. No simple combined coefficient of heat transfer has been found to express these two processes. Coefficients based on wet bulb temperature difference or even on over-all enthalpy difference between the refrigerant temperature and air wet bulb temperature give only approximate results.

Heat transfer in the evaporative condenser has been discussed by Goodman,[1] Thomsen,[2] James[3] and Wile.[4] The following analysis is from the latter reference.

Heat transfer from the refrigerant vapor through the tube wall and the water film to the water surface can be expressed by

$$U = \frac{q}{A(t_c - t_s)} \qquad (1)$$

where

U = coefficient of heat transfer from refrigerant vapor through the tube wall and water film to wetted surface, Btu per hr F sq ft

q = heat input to the condenser, Btu per hr

A = surface area of the condenser tubes, sq ft

t_c = condensing temperature, F

t_s = the wetted surface temperature, F

Heat transfer from the wetted surface to the air can be expressed conveniently in the form of a by-pass factor Z:

$$Z = \frac{h_s - h_2}{h_s - h_1} \tag{2}$$

where

h_s = enthalpy of saturated air at the water film temperature, Btu

h_2 = enthalpy of the leaving air stream, Btu

h_1 = enthalpy of the entering air stream, Btu

By the use of equations 1 and 2 the coefficient U and by-pass factor Z can readily be obtained from test data. They provide a means for predicting performance under various conditions of operation within the range of the test data.

Figure 3 shows the results of tests on two condensers of the same physical size but constructed respectively of 3/4 in. steel pipe and 5/8 in. OD copper tube. Both condensers had 20.6 sq ft of prime surface, 3.43 gpm of recirculated water and 517 cfm of air per sq ft of face area.

It is highly important to note that the data in Figure 3 applies only to a condenser of the exact construction and characteristics of the test unit. The heat transfer characteristics of the condenser are influenced considerably by the number of rows deep in the direction of air flow, the rate of water recirculation and the air flow rate.

It should be understood that the actual heat transfer process in an evaporative condenser is a highly complex one and that the surface temperature as derived from Equation 2 can be considered at best an empirical approximation of the actual mean surface temperature. Likewise, the coefficient U of Equation 1 must be regarded as empirical rather than corresponding to the physical value of conductance between the surface and refrigerant. For these reasons, it cannot be considered safe practice to extrapolate test data appreciably beyond the range of the tests, nor to other coil depths or physical construction.

Rating Curves

Ratings of evaporative condensers are generally expressed in the form of tables or curves showing either the refrigeration load or the actual condenser heat input. **Figure 4** shows the variation in condenser rating (refrigeration load) with changes in condensing and air wet bulb temperature. The curves are based upon actual test results using equations 1 and 2 to inter-

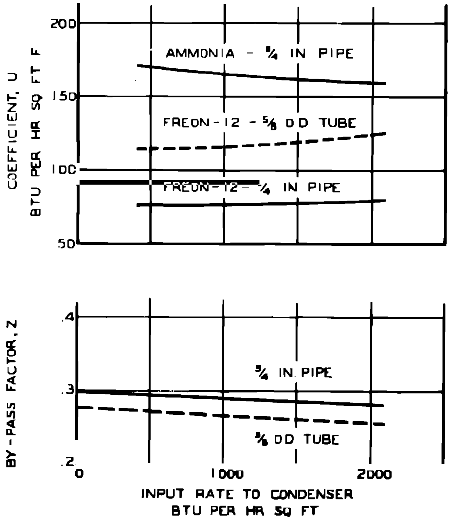

Fig. 3. Comparative Condenser Test Results

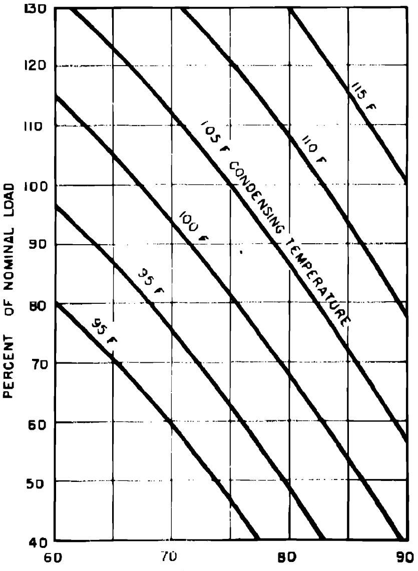

Fig. 4. Typical Rating Curves

polate within the range of test data. It should again be emphasized that the data in Figure 4 applies to a particular construction and not to condensers in general.

Figure 4 is based upon an evaporating temperature of 40 F and includes the ratio of condenser load to refrigeration load as given in Figure 1, chapter 24. The effect of evaporating temperature on the condenser load is slightly different at high condensing temperatures than at low condensing temperatures but an average can be obtained that will represent true performance within a very small error. The figures in **Table 1** are based on such an average.

Table 1. Freon-12 Load Factors for Various Evaporating Temperatures

Evap. Temp, F	Load multiplier
−20	1.22
−10	1.17
0	1.13
10	1.09
20	1.06
30	1.03
40	1.00
50	0.98

Standard Rating Conditions

The standard rating conditions[5] that have been accepted generally by the industry[6] are:

Barometric pressure, 29.92 inches of mercury
Entering air dry bulb, 90 F
Entering air wet bulb, 75 F
Ambient air dry bulb, 90 F
Water entering unit, 75 F
Saturated temperature of refrigerant vapor entering unit, 105 F
Actual temperature of refrigerant entering unit, 125 F minimum
Ratio of net refrigerant heat rejection effect to the net refrigeration effect, 1.20

Actual operating conditions, especially wet bulb temperatures, may vary considerably from standard conditions. It is also common practice to select ammonia condensers for lower than standard condensing temperatures, 95 F being a rather common selection point.

Effect of Air and Water Quantities

The amount of surface used by various manufacturers to obtain the same tonnage rating varies over a fairly wide range and is strongly affected by the amounts of air and water that are circulated, as well as the tube size, spacing and other physical characteristics.

Maximum air quantities are generally limited by fan motor horsepower considerations and the maximum velocity that can be tolerated through the eliminators without carry-over of water particles.

Condenser capacity is affected only slightly by increasing water circulation above a minimum required rate. In one series of tests an increase of water flow of 35 percent increased the capacity only 4 percent. Ample water circulation well distributed over the tubes is essential to minimize scale formation.

Desuperheating Coils

The capacity of a Freon 12 evaporative condenser may be increased by the order of 15% and of an ammonia evaporative condenser by the order of 20% by locating additional heat transfer surface in the form of an extended fin coil through which the superheated vapor from the compressor is fed on the way to the condenser tube bundle. The desirability of these desuperheating coils for increasing the capacity of the condensers can be determined by an economic balance between the increased cost and the increased capacity. Claims have also been made that the reduction in hot gas temperature greatly reduces the scale formation.

While it is true that the desuperheating coil reduces the temperature of the hot gas it does not necessarily follow that this results in any appreciable decrease in the temperature of the tube itself. Heat transfer from a dry gas to the tube wall is poor as compared to the heat transfer from a condensing vapor to the tube wall. The actual tube temperature depends upon the relation between the rate at which heat is conducted from the vapor to the tube and the rate at which it is conducted away from the outside of the tube through the medium of the circulating water and air flow. Under conditions of normal operation a very considerable amount of superheat causes little change in the tube temperature.

An important use of desuperheating coils on ammonia compressors is to improve the effectiveness of oil separators usually placed in the compressor discharge line. By allowing the hot gas to pass through the desuperheating coil first and then into the oil separator part of the oil vapors are condensed to a liquid before entering the oil separator. It is obviously important that the desuperheat coil not be oversized to the extent where condensation of refrigerant takes place as this would produce the undesirable condition of supplying liquid refrigerant to the oil separator.

Condenser Location

When evaporative condensers are located indoors they can be used to aid in ventilating the machine room or may be connected by duct work to the outside. The indoor location is often required due to lack of space outdoors or the necessity of avoiding long runs of refrigerant piping to the condenser.

Evaporative condensers are often located outdoors such as on roofs and can be protected from freezing during cold weather by locating the pump and a water tank in a heated area below the condenser. The piping can be arranged so that each time the pump stops all of the water drains back into the water tank and thus avoids freezing. Condensers may also be obtained with air dampers to reduce the air flow during cold weather so as to avoid freeze-up and to maintain reasonably high head pressure.

Capacity Modulation

To assure the proper operation of expansion valves or other refrigerant control devices it is often necessary to avoid extremely low condensing pressures such as might result during winter operation with an evaporative condenser. Capacity reduction, under the control of a pressure switch, can be obtained in several ways by: (1) intermittent operation of the water circulating pump, (2) intermittent operation of the blowers, (3) intermittent operation of both the pump and air blowers, (4) placing a modulating or two position damper in the air stream to reduce the air flow. Of these various methods, item 1, the intermittent operation of the water pump, is definitely not recommended since it causes a thin film of scale to be deposited each time the pump is stopped and the water is evaporated from the tube surface. After continued cycling a thick deposit of scale will result. Of the other three methods the use of a modulating or two position damper is often most desirable since it may be used to prevent freezing in cold weather. For cold weather operations the pump and a sump tank should be located in a heated area as mentioned above.

Evaporation of Refrigerant from Receiver

It is not uncommon in connection with the use of evaporative condensers for the liquid receiver to be located in an ambient temperature that is higher than the condensing temperature. During continuous operation, flow of liquid into the receiver will maintain its temperature at a satisfactory level. In order to avoid excessive receiver temperatures the receiver should always be protected from direct sunlight and in some instances it would be desirable to insulate the receiver from the surrounding air temperature.

During a period when the system is shut down it often happens that a condenser located on the roof is subjected to higher temperatures than the compressors, with the result that refrigerant may evaporate from the receiver or the condenser and collect in the compressor head and the discharge piping. Where such conditions are likely to occur it may be desirable to install a check valve in the hot gas line where it enters the condenser.

Sub-Cooling Coils

Improvement in system efficiency can be obtained by sub-cooling the liquid refrigerant below its corresponding condensing temperature. With Freon 12 refrigerant, in air conditioning or the usual commercial installations, the system efficiency will be improved approximately 1/2 percent for each degree that the liquid is cooled. Liquid sub-cooling can be accomplished with evaporative condensers by placing an additional coil underneath the tube bundle where it will be in the unheated inlet air. While the amount of sub-cooling possible will depend on operating conditions it will seldom exceed 15 F or 20 F with Freon 12 refrigerant. It should be noted that the sub-cooling coil improves the capacity and the efficiency of the entire system because it increases the refrigeration effect without increasing the compressor load.

With ammonia the heat of the liquid is so small in comparison to the total refrigeration effect that sub-cooling coils are rarely used except for extremely low temperature applications.

Dry Coil Operation

It is sometimes proposed to operate an evaporative condenser without water during the winter season. The reduction in capacity for the dry coil is usually so great that the full rated load could not be carried even under extremely low temperature conditions. If a desuperheating coil is used the considerable increase in surface improves this condition but would seldom permit operating under full load even in the coldest weather. The ratio of dry coil capacity to normal capacity varies with the design of the condenser and can usually be obtained from data supplied by the manufacturer.

Purge Valve

Purge valves should always be located on the receiver and this is particularly important with evaporative condensers. It is best to locate the purge valve at the opposite end of the receiver from the connection to the condenser. This will insure complete purging of air from the system with minimum loss of refrigerant vapor. With the purge valve located on the receiver it is not necessary to purge other points in the system.

Receivers should be located below the discharge connection from the condenser, using large enough pipe to avoid liquid traps or gas binding.

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26. EVAPORATORS

THE cooling effect of a refrigerating system takes place in the evaporator, where heat from the medium to be cooled (air, water or brine) vaporizes the liquid refrigerant. As the refrigerant boils away, it is constantly replaced by more liquid from the receiver, while the vapor is returned through the suction piping to the compressor.

This chapter covers various types of ammonia and Freon evaporators in common use for industrial and air conditioning applications.

Good evaporator design is concerned with (a) the production of high **heat transfer,** and (b) sound **practical design** to give long life and freedom from functional troubles such as liquid carry-over, oil logging, and scaling.

The amount of heat absorbed in an evaporator is shown by the following two equations:

$$Q = wc(t_1 - t_2) \qquad (1)$$

where

Q = Btu per hr, given up by the medium
w = lb of medium circulated, lb per hr
c = specific heat of medium, Btu per lb F (0.24 for air, 1.0 for water, 0.7 for C_aCl_2 brine of 1.2 sp gr)
t_1 = entering temperature, F of medium being cooled
t_2 = leaving temperature, F of medium being cooled

$$Q = UAD \qquad (2)$$

where

Q = heat transferred or given up, Btu per hr
U = overall heat transfer coefficient, Btu per sq ft hr F, based on outside surface
D = logarithmic mean temperature difference, F
A = area of heat transfer surface, sq ft (outside)

Equations (1) and (2) may be combined as below:

$$UAD = wc(t_1 - t_2)$$

$$A = \frac{wc(t_1 - t_2)}{UD}$$

Once the factor U is established, the performance of the cooler may be predicted. U varies materially, however, with design of evaporator, the fluid and its velocity, D, and with fouling of tube surface.

Heat Transfer

Overall heat transfer through a tube is a function of (a) film coefficient on refrigerant side, (b) tube-wall conductivity, (c) film coefficient on fluid side, and (d) ratio of outside to inside tube surface, and is evaluated in the following expression:

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{h_{fs}} + \frac{L}{K} + \frac{R}{h_{ws}} + \frac{R}{h_w} \qquad (3)$$

where

U = overall heat transfer factor, Btu per sq ft hr F
h_f = film transfer coefficient, Freon side, Btu per sq ft hr F
h_{fs} = scale coefficient; Freon side, Btu per sq ft hr F
L/K = tube-wall resistance, wall thickness divided by conductivity; wall thickness in feet conductivity in Btu ft per sq ft hr F
R = Ratio of outside total surface to inside tube surface
h_{ws} = scale coefficient, water side
h_w = film coefficient, water side

The **refrigerant side coefficient** depends on thorough wetting of the surface, rapidity of boiling and rate of escape of gas bubbles from tube surface. As gas bubbles form from the boiling action, they tend to cling to the surface of the tube, reducing the effective surface exposed to liquid. The rougher the surface, the smaller the bubbles, and the more quickly they are released. Thus, in flooded coolers, heat transfer is greatest for small tubes and those with rough surfaces (**Fig. 1**).

An increase in mean temperature difference, *D*, will cause greater turbulence in boiling, sweeping away of the gas bubbles, and improved heat transfer, as shown in Fig. 1. Close spacing of tubes will also increase boiling turbulence. A special type of low extruded fin on a small diameter tube has given very high heat transfer rates; due to the closely nested surface, the turbulence actually becomes foaming and results in high percentage of wetted surface with relatively small amounts of refrigerant. The refrigerant side coefficient is better when:

1. Tubes are in liquid refrigerant contact, or submerged
2. Tubes are small
3. Tubes have roughened or extruded surface to refrigerant
4. Tubes are closely spaced for maximum turbulence of boiling
5. *D* is high, say 12 F or more, and
6. Oil is absent.

The **resistance** of a clean **tube-wall** to **heat flow** is usually so low as to have little effect on the overall heat transfer. Fouling of the tube-wall surface, particularly on the side of the fluid, can materially decrease transfer. Steel tubes foul more readily than copper. Higher fluid velocities decrease fouling. For small copper tubes, good design practice is to use a fouling factor of 2,000, h_{ws} in Equation (3).

For a given fluid, the **film coefficient on the fluid side** is higher when:

1. Fluid velocity or turbulence is high
2. Viscosity is low and
3. Tubes are small.

Most applications deal with turbulent flow where the Reynolds number is above 3,000 (see Chap. 6) as differentiated from streamline flow (Reynolds number below 2,500).

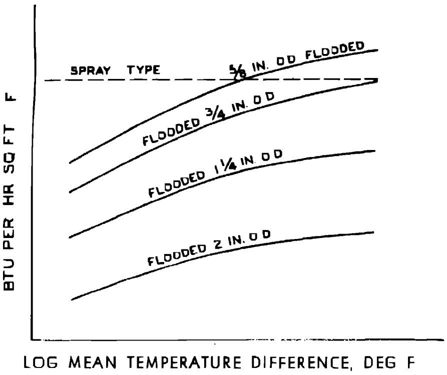

Fig. 1. Heat Transfer Affected by Tube Size and Surface

The effect of varying the **ratio of outside to inside tube surface** can be seen by referring to Equation (3), where the overall factor *U* is based on outside surface. If fluid side and refrigerant side coefficients are about equal, bare tubes are normally used. With high fluid velocities up to 8 fps, the fluid side coefficient may become much higher than the refrigerant side coefficient. In this case, it is economical to utilize a finned tube to increase outer tube surface. The finning ratio should be only sufficient to derive maximum benefit from the high fluid side coefficient. For example, in Equation (3), if $h_f = 225$; $L/k = .0000183$; $h_{fs} = 5{,}000$; $h_{ws} = 2{,}000$ and $h_w = 1{,}000$, first take *R* as 10 to 1, secondly as 4 to 1. For $R = 10$ to 1, $U = 51$. For $R = 4$ to 1, $U = 95$. It will be seen that in the foregoing conditions, the *U* factor almost doubles when ratio of outside to inside surface is reduced from 10 to 1 to 4 to 1. Therefore, the mention of total surface is no measure of cooler performance unless both quality of surface and ratio of outside to inside surface are known.

Table 1 shows the range of values of overall heat transfer for some of the commonly used types of evaporators. The top limits are for high *D* values of 12 F or more and for turbulent fluid flow. Fluid velocities for bare tubes range up to 5 fps and for finned tubes up to 8 fps.

Table 1. Overall Heat Transfer

Current conservative design values of heat transfer coefficient U for bare tube coolers, unless mentioned otherwise, are as follows:

	Min	Max
Flooded shell-and-tube cooler (water to ammonia or Freon)	50	150
Flooded shell-and-finned tube high velocity Freon water cooler	30	150
Flooded shell-and-tube cooler (brine to ammonia)	45	100
Flooded shell-and-tube cooler (brine to Freon)	30	90
Dry expansion shell-and-tube cooler, Freon in tubes, water in shell	50	115
Baudelot cooler, flooded (ammonia or Freon to water)	100	200
Baudelot cooler, dry expansion (ammonia to water)	60	150
Baudelot cooler, dry expansion (Freon to water)	60	120
Double-pipe cooler (water to ammonia)	50	150
Double-pipe cooler (brine to ammonia)	50	125
Shell-and-coil cooler (water to ammonia)	10	25
Shell-and-coil cooler (water to Freon)	10	25
Spray type shell-and-tube water coolers (ammonia or Freon)	150	250
Tank-and-agitator, coil type water cooler, ammonia, flooded	80	125
Tank-and-agitator, coil type water cooler, Freon flooded	60	100
Tank, ammonia, brine cooling, coils between can in ice tank	15	40
Tank, high velocity raceway type, brine to ammonia	80	110

Actual **design values** of heat transfer, accepted by the industry as a basis for guaranteeing brine coolers, are given in **Figs. 2** and **3**. These charts clearly indicate the effect on heat transfer of tube size, mean temperature difference D, brine velocity, and brine viscosity with lower brine temperatures and heavier brine densities. Figs. 2 and 3 are based on a brine concentration which will freeze at a temperature approximately 20 deg F below the average brine temperature. These curves may be used conservatively for Freon-12 by applying a correction factor of 0.75 to the ammonia performance, based on 1¼-in. or smaller tubes for Freon.

Static head correction. In rating flooded brine coolers, the effect of static head is often neglected. The pressure at the bottom of the shell may be a psi or more greater than at the top, with a resultant higher refrigerant boiling temperature and less capacity. For low mean temperature difference and low evaporating temperatures in the larger size coolers, the capacity penalty is appreciable. In **Table 2** is given the liquid static head correction in degrees F to be applied to log MTD in Figs. 2 and 3.

Table 3, for flooded bare copper tube Freon-12 water coolers, gives one manufacturer's performance guarantee. Experience has shown it to be conservative for both large and small tonnage applications and it is representative of normal practice for bare tubes. Heat transfer rates up to 200 Btu per sq ft hr F could be realized with higher water velocities and greater tube submergence.

Flooded Shell-and-Tube Ammonia Brine Cooler

This type of cooler, **Fig. 4**, has long been the standard for industrial brine cooling applications. The construction is all-steel with either 1¼-in. or 2-in. tubes expanded into the tube sheets. Typical specifications are given in **Table 4**.

Tube heads are arranged for multipass brine flow through the tubes. The perform-

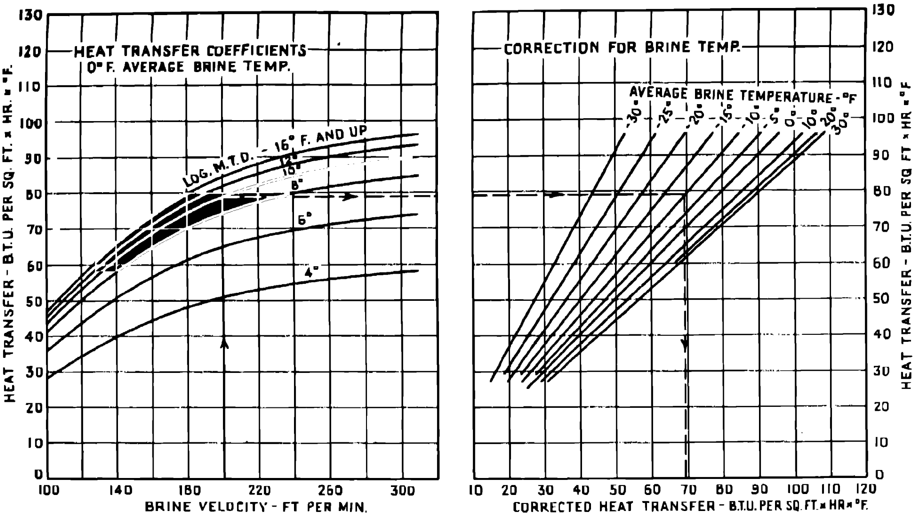

Fig. 2. Heat Transfer from Ammonia to Brine in Flooded Coolers—2-in. tubes

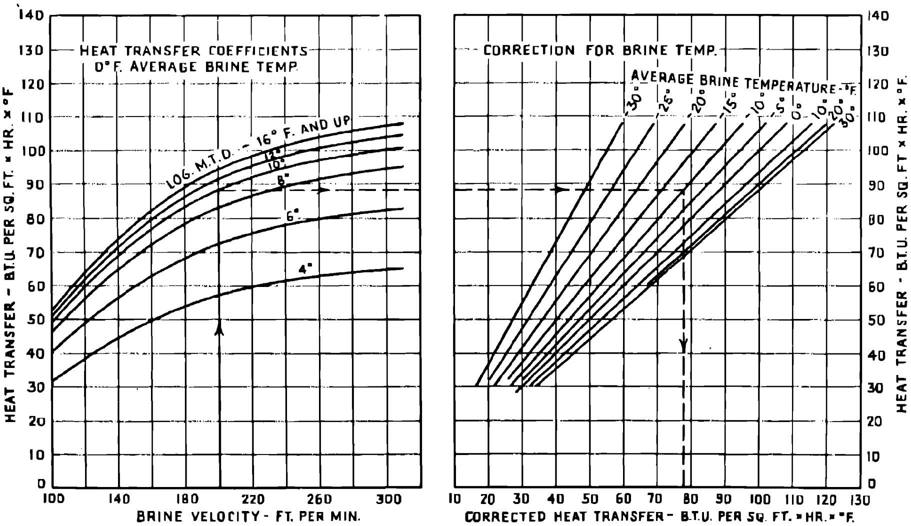

Fig. 3. Heat Transfer from Ammonia to Brine in Flooded Coolers—1¼-in. tubes

Example: For average field operating conditions with a flooded cooler having 1¼-in. plain steel tubes, find the heat transfer coefficient when −10 F average brine is circulated at a velocity of 200 fpm and the *D* is 10 F. Following the dashed line and arrow, proceed vertically up from the brine velocity of 200 fpm to the transfer curve for 10 F *D*, go horizontally to the right to the correction curve for −10 F average brine, and, reading vertically down, obtain the corrected heat transfer coefficient of 77.7 Btu per sq ft hr F.

For identical conditions, except with 2-in. tubes, the transfer coefficient is 69.5 Btu per sq ft hr F.

Table 2. Liquid Static Head Correction in deg F for Various Evaporator Temperatures, Horizontal Flooded-Type Ammonia Brine Coolers

Shell diam, in.	10 in. vac −42.1 F	5 in. vac −34.5 F	0 −28 F	5 psi −17.2 F	10 psi −8.4 F	15 psi −1.0 F	20 psi 5.5 F	25 psi 11.3 F	30 psi 16.6 F	40 psi 25.8 F	45 psi 30.0 F
16	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.15	0.15	0.15	0.15
20	0.4	0.4	0.3	0.3	0.3	0.3	0.3	0.2	0.2	0.2	0.2
24	0.7	0.7	0.5	0.4	0.4	0.4	0.35	0.25	0.25	0.2	0.2
30	0.9	0.9	0.7	0.5	0.5	0.5	0.4	0.3	0.3	0.25	0.25
34	1.1	1.0	0.8	0.6	0.6	0.55	0.45	0.35	0.35	0.3	0.3
38	1.3	1.1	0.9	0.7	0.7	0.6	0.5	0.4	0.4	0.3	0.3
42	1.4	1.2	1.0	0.75	0.75	0.65	0.55	0.45	0.45	0.35	0.35
45	1.5	1.3	1.0	0.8	0.8	0.65	0.55	0.45	0.45	0.35	0.35
50	1.6	1.4	1.1	0.9	0.9	0.7	0.6	0.5	0.5	0.4	0.4
54	1.7	1.5	1.2	1.0	1.0	0.75	0.65	0.55	0.55	0.4	0.4
60	1.9	1.7	1.4	1.1	1.1	0.85	0.75	0.6	0.6	0.5	0.5

ance of this cooler, Figs. 2 and 3, is based on use of calcium chloride brine with the shell flooded with liquid ammonia to a static height of 0.8 of the internal shell diameter. Brine temperature differences, entering minus leaving, in practice run from about 4 to 15 deg F. **Table 5** compares flow of water to $CaCl_2$ brine of 1.2 sp gr for various temperature differences.

Consley[5] has discussed the use of swirl strips in the tubes to increase brine turbulence at low velocity. Although the gain in heat transfer is somewhat balanced by increased pressure drop, swirls may be used to increase capacity of existing equipment in the field, as in **Fig. 5.**

Flooded Shell-and-Tube Freon Coolers

This cooler has found its greatest field of application in chilling water for air conditioning service. Recent installations have been made, however, in large tonnage capacities for industrial brine cooling. Usual construction is steel shell and copper tubes. Latest trend is to the use of finned tubes and higher fluid velocities.

Table 3. Overall Heat Transfer Factors in Flooded Coolers

(Btu per sq ft hr F)

Water velocity, fpm	Mean temperature diff, deg F				
	6	8	10	12	15
150	67	76	83	90	97
200	83	95	103	110	118
250	97	109	115	122	529
300	103	115	123	130	138

These factors are based upon the actual temperature of refrigerant in the shell, and a Freon level of approximately [illegible] of the total height, with commercially pure, oil-free Freon and [illegible]-in OD copper tubes.

For $CaCl_2$ brine cooling, construction is generally of steel throughout, although in plants where brine is chemically treated and kept under supervision, non-ferrous alloys are coming into use. Usual practice is to groove or serrate the tube holes in the tube sheet, of either steel or non-ferous alloy, then expand the tube to a tight fit in the tube hole. Some coolers are made with light copper or alloy tube sheets with tubes silver soldered to the tube sheet.

A typical design of bare tube cooler is shown in **Fig. 6** and its hook-up with float control in **Fig. 7.**

In this cooler the suction is taken off the top at several points into a main header containing superheater surface with the compressor connection in the center. This type of design uses a somewhat smaller shell, the shell itself being almost filled with tubes on close spacing. The need for distributing headers on the liquid side is mainly to cause uniform turbulence through the length of the shell. For typical specifications of this type cooler, see **Table 6.**

A typical design of finned tube cooler is shown in **Fig. 8** and its hook-up with pilot thermal valve control in **Fig. 9.** Here the super-heater surface is located within the shell. The close nesting of finned tubes

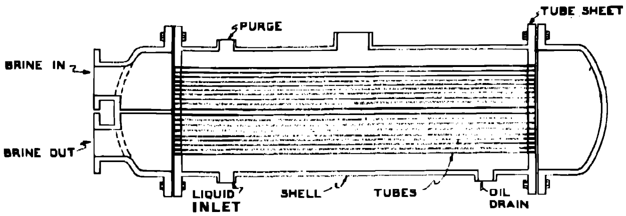

Fig. 4. Ammonia Shell-and-Tube Brine Cooler

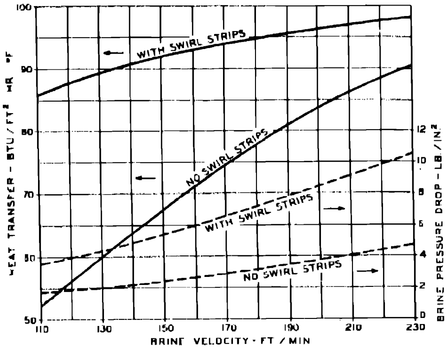

Fig. 5. Effect of Swirl Strips—Ammonia Brine Coolers

concentrates heat transfer surface to produce violent turbulence and foaming, which effect uniform wetting of the entire surface. Thermal valve action causes some carry-over of foam containing oil. This foam is dried out by the superheater and the oil returned to the compressor via the suction line. Specifications for this cooler are as in **Table 7.** The ratio of outside to inside surface is approximately 4 to 1.

Spray-Type Coolers

This type is being largely superseded by flooded coolers except for applications at very low temperatures where the static head penalty on flooded coolers is too great or at low MTD's where the performance of flooded coolers falls off rapidly. Since the wetting of the tube surface by sprays is uniform regardless of heat load, heat transfer of spray-type coolers is independent of MTD.

Construction is similar to flooded coolers except for location of a spray header with nozzles above the tube bundle. The Freon or ammonia is collected in a sump at the bottom of the cooler and recirculated by a low-head pump through the spray system, flowing down over the tubes. Recirculating flow ratio is high compared to evaporated refrigerant flow, assuring thorough wetting of tubes. Refrigerant charge is low and is much less affected by load variations than the flooded coolers. Greatest disadvantage of spray coolers is necessity of mechanical pump and high installation cost.

Table 4. Specifications of Typical Ammonia Flooded Brine Coolers with 1¼-in. Tubes

Shell diam, in.	Length, ft	No. tubes	Surface area, sq ft	Weight, lb	Ammonia charge, lb
16	16	60	315	2,700	420
20	16	103	538	4,400	642
24	16	163	853	6,500	868
30	16	247	1,292	9,200	1,427
34	16	341	1,784	12,200	1,880
38	16	426	2,229	15,400	2,320
42	16	530	2,773	18,500	2,795
45	16	612	3,202	21,500	3,140
50	16	764	3,997	26,500	3,830
54	16	868	4,541	30,200	4,340
60	16	1,082	5,661	37,700	5,450

Table 5. Rate of Flow of Fluid at Differences in Temperature to Equal One Ton of Refrigeration

Temperature change, deg F	Water circulated, gpm/ton	$CaCl_2$ circulated, gpm/ton
1	24	28.6
2	12	14.3
4	6	7.15
6	4	4.8
8	3	3.6
10	2.4	2.86
12	2	2.4
14	1.72	2.04
16	1.5	1.8
18	1.33	1.6
20	1.2	1.43

Non-Flooded Shell-and-Tube Freon Coolers

The so-called dry expansion cooler (Fig. 10) in which the refrigerant is in the tube, is becoming more popular for small and medium tonnage applications, say up to 150 tons per unit. Thermal valve is used for Freon control. Since the valve requires superheat for its operation, this cooler is not good for applications requiring close temperature splits between Freon and fluid, unless used in conjunction with heat exchanger. There are two main types of design, first the U-tube type, in which the entire tube bundle is removable from the shell and, second, the fixed-tube type which is like an ordinary cooler except that

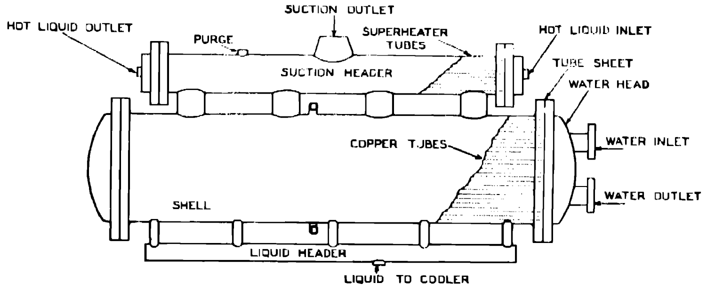

Fig. 6. Typical Freon Flooded Water Cooler

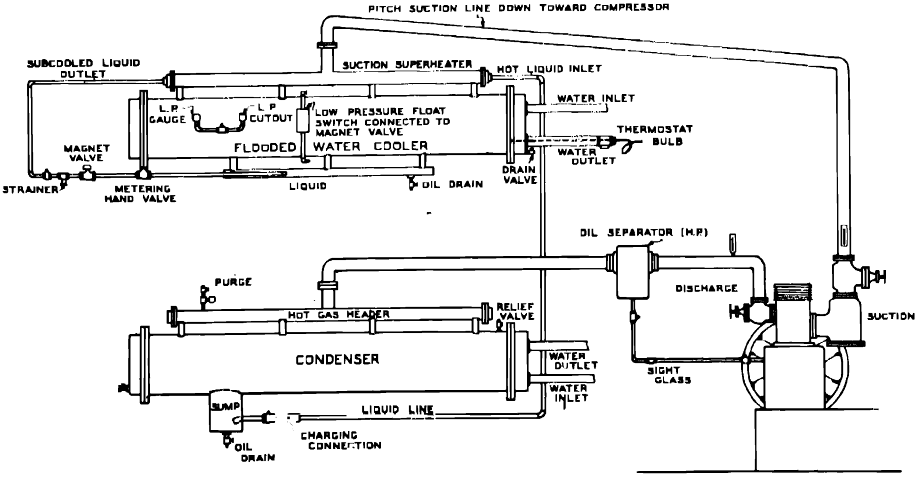

Fig. 7. Recommended Hook-up of Freon Water Cooler

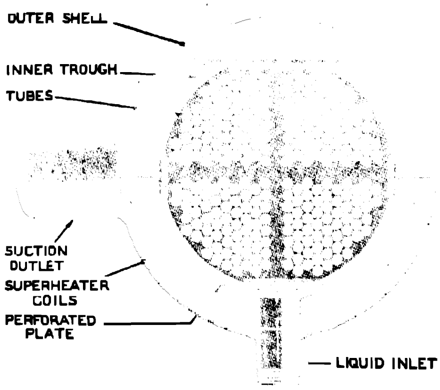

Fig. 8. Finned Tube Freon Water Cooler

it has shell side baffles. The U-tube type is shown.

The advantages of this cooler are that the water friction is low, the control is simple with the thermal expansion valve, Freon charge is small, and there is no difficulty with oil return since the feed is positive. This cooler does not readily freeze or burst tubes, and for this reason, lower Freon temperatures can be used. The overall heat transfer is lower than with flooded coolers, but generally the actual installed cost of the dry cooler is about the same as the flooded. The figures which follow indicate design factors used by one manufacturer, based on a constant water velocity:

Table 6. Specifications of Typical Freon Flooded Water Coolers with Bare Copper Tubes, ⅝-in. OD

Shell diam, in.	Length, ft	No. tubes	Surface area, sq ft	Weight, lb	Normal capacity, tons 10° MTD and 5 fps vel	Freon charge, lb
10¾	5	91	74	500	7.4	75
	10	91	148	950	14.8	150
12¾	10	133	216	1,150	21.6	200
	16	133	345	1,840	34.5	320
14	14	166	380	1,600	38.0	345
16	14	220	504	2,200	50.4	445
18	16	306	800	3,360	80.0	620
20	16	364	951	4,240	95.1	800
24	16	566	1,480	6,000	148.0	1,080

Table 7. Specifications of Typical Freon Flooded Water Coolers with Finned Copper Tubes, ⅝-in., 16 Fins per in.

Steel diam, in.	Length, ft	No. tubes	Surface area, sq ft	Weight, lb	Normal capacity, tons 10° MTD 8 fps vel	Freon charge, lb
16	12	52	320	1,530	32	236
20	12	108	665	2,225	66	520
24	12	200	1,235	3.210	124	919
30	12	338	2,080	4,660	208	1,590
36	12	564	3,475	7,215	348	2,620
42	12	764	4,700	9,940	470	3,600
50	12	1,250	7,700	14,315	770	5,900

D *deg F*	*U* *Btu per sq ft hr F*
8	80
10	105
12	110
15	115

Other manufacturers have based their ratings on varying water velocities rather than mean temperature difference.

The straight tube design, while having the disadvantage of lack of access to the water side, has the advantage that it may be re-tubed if necessary. Re-tubing, however, is seldom necessary with Freon in the tubes, since there is little likelihood of rupture due to freezing. From the practical standpoint, the removability of the U-tube bundle is not too great an advantage.

Shell-and-Coil Coolers

The shell-and-coil cooler has its application chiefly in drinking water systems, for bakeries and photographic laboratories. It consists of a steel or alloy tank with a circular coil in which the refrigerant is circulated. The water or brine is in the tank outside the coil, as in **Fig. 11**. This is generally a dry expansion application, for the reason that the resistance to heat transfer is so much greater on the water side than on the Freon side, that there is little point to going to further complication than the simple thermal expansion valve. This type of cooler has the advantage of flywheel effect, making it popular for applications where peak loads are high and infrequent.

There are a number of specialty applications of small water coolers for the same purpose as the shell-and-coil, in two of

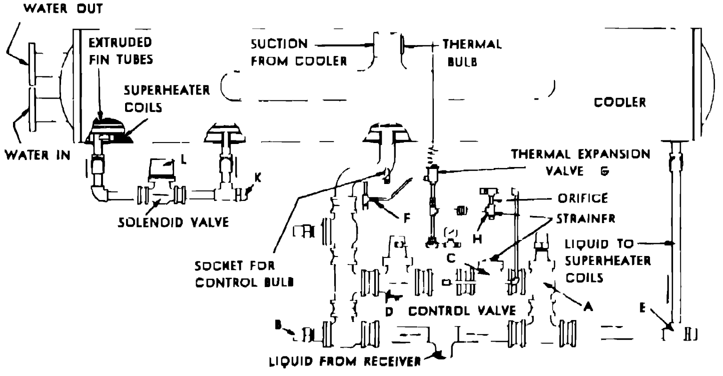

Fig. 9. Typical Thermal Valve Hook-up of Freon Water Cooler

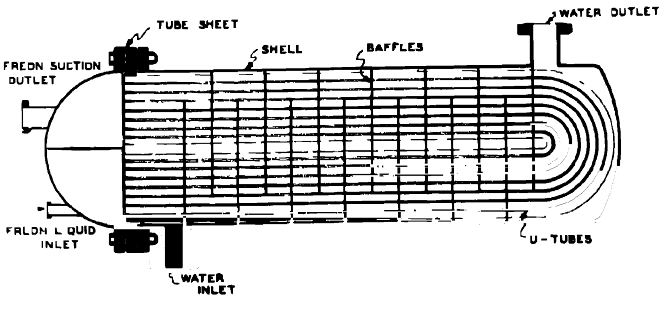

Fig. 10. Dry Expansion Freon Water Cooler

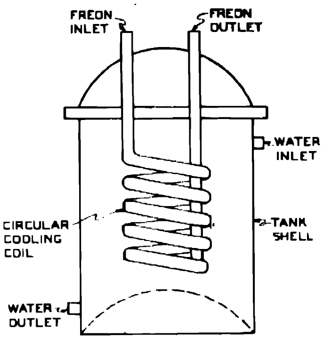

Fig. 11. Shell-and-Coil Cooler

which an attempt is made to obtain the efficiency of a high-velocity evaporator combined with the holdover effect of a tank cooler. This has the disadvantage that once the incoming water has passed the high-velocity evaporator, it must have been cooled, since it is not recirculated. The arrangement does not allow the continuous cooling at low average rate as in the simple pressure tank or shell-and-coil cooler. Also, the high-velocity cooler can freeze and cause damage to the coil, whereas this is very unlikely with the shell-and-coil type. Due to icing, these coolers should not be used where it is desired to cool water below about 38 F.

The Baudelot Cooler

There are many industrial applications requiring water chilled to 35 F, sometimes as low as 33 F. Shell-and-tube or shell-and-coil coolers are not suitable for this service as they will ice up. The Baudelot cooler of **Fig. 12** lends itself very well to producing near-freezing temperatures. Although it, too, can ice up, no damage can be done, and since it is open to visual inspection, ice build-up can be controlled. It consists of a series of pipes one under the other, through which refrigerant circulates and over which the water to be cooled trickles down in a thin film from an overhead pan in which small distributing holes are drilled. A more modern design comprises stamped corrugated stainless steel sheets in which the corrugations form channels for the refrigerant flow. The sheets are highly polished and form a sanitary, easily cleaned surface. Also, they provide a continuous surface offering better control of water distribution.

Ammonia type is usually flooded, using the conventional gravity feed system with surge drum and low pressure float. Freon type is usually either dry expansion or combination dry and flooded.

The double-pipe cooler is not used very widely in modern practice for other than a few specific applications. It is used in the

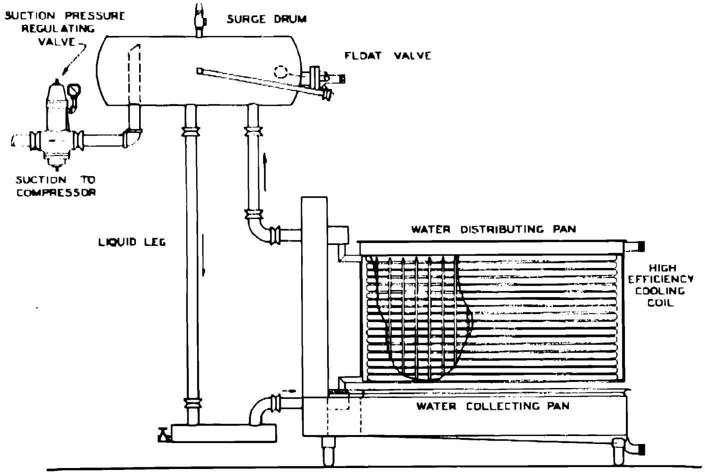

Fig. 12. Baudelot Cooler

cooling of wine and beer during manufacture and in the oil refining industry for oil chilling. The cooler is efficient, being adaptable to counterflow, but takes up more space than the shell-and-tube type, has more refrigerant side friction and is not readily cleanable. It is virtually obsolete for cooling of water or brine.

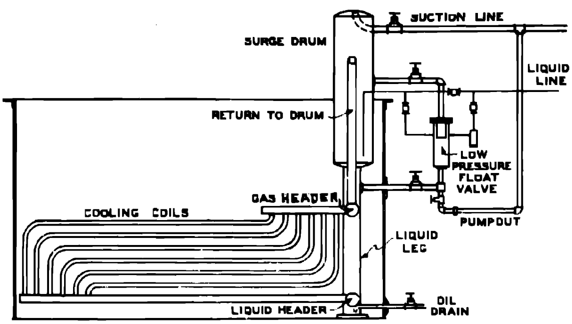

Fig. 13. Typical Boiler Type Coil with Flooded Control

Tank-Type Cooling Systems

The tank or boiler-type coil system (**Fig. 13**) for cooling brine or water has been in almost universal use for large systems for many years, where sanitation is secondary. A large steel tank contains the fluid to be cooled. At one side or the center of the tank is placed a closely nested set of pipe coils, baffled off from the tank proper and fitted with an agitator to circulate the brine or water in the tank over the coils at a relatively high velocity (100 to 150 fpm).

The agitator circulates large quantities of brine or water to maintain uniform temperatures throughout the tank with total temperature difference less than 1 deg F. Coils are usually arranged with short length of travel from liquid to suction header to minimize refrigerant pressure drop. In an alternate arrangement to the one shown, the coils are bent in V-form between the headers.

This same type of system is used for freezing ice in can-type tanks. The tank is enlarged to accommodate the ice cans with brine flowing past the cans at approximately 25 fpm.

Typical performance of these coils is given in **Table 8**. For flooded ammonia operation, correction must be made for static height of liquid from liquid level in surge drum to liquid header.

Some ice-making tanks still use flat coils between rows of cans, usually controlled by hand expansion. Heat transfer values for these coils are given in Table 1.

Brines

Calcium chloride brine is used for most industrial refrigeration jobs. Sodium chloride is used in open spray systems where contact of vapor with food is involved. Its heat transfer characteristics are very similar to calcium chloride. Calcium chloride is seldom used below −30 F because of poor heat transfer due to its increasing viscosity. For very low temperatures, methanol and chlorohydrocarbon liquids can be used. Two of these which have been used in recent years to temperatures as low as −100 F are methylene chloride and trichlorethylene. Some of their properties are shown in **Table 9**.

Oil and Heat Exchangers

Since the presence of oil in evaporators, although unavoidable, interferes with performance, means should be provided to reduce the quantity as much as possible. In ammonia coolers the oil, being heavier than the refrigerant, usually settles out at low spots from which it can be drained.

With Freon-12, oil raises the boiling point one degree for each 5% of oil present. Also, oil is miscible with liquid Freon-

Table 8. Typical Performance of Boiler Type Coils (High Velocity)

(Overall heat transfer, Btu per sq ft hr deg F)

Velocity, fpm	Water	Brine			
		30 F	15 F	0 F	−15 F
50	68	52	47	42	38
75	88	72	66	60	53
100	110	94	85	77	68
125	132	116	105	95	84
150	150	133	121	109	97

Table 9. Properties of Chlorohydrocarbon Liquids

	Methylene chloride	Trichlorethylene
Freezing pt, F	−142.1	−126.4
Sp ht at −22 F, Btu/lb/F	0.275	0.222
Sp gr at −22 F	1.42	1.55
Viscosity, centipoises at −4 F	.676	.90

12 and cannot be drained to atmosphere. Various recovery devices are available to "still" some of the oil and return it to the compressor, thus maintaining a tolerable oil concentration (not more than 10%) in the cooler. For float-controlled flooded coolers, the oil still is commonly used. It is a form of double-pipe heat exchanger using liquid to the cooler as a source of heat. A sample of liquid in the cooler is passed through the still, the liquid evaporated and the remaining oil returned to the compressor.

Suction-line heat exchangers are used with Freon-12 evaporators for a two-fold purpose: (a) to assist in oil return, and (b) to serve as liquid precoolers. As oil return devices, they apply particularly to thermal expansion valve fed evaporators, where the superheat required to operate the expansion valve is produced in the heat exchanger with warm liquid as the source of heat. This action forces some liquid-oil mixture spill-over from the evaporator. The exchanger evaporates the liquid and permits the oil to return to the compressor via the suction line. Used as liquid precoolers, heat exchangers can show some gain in overall system capacity. They have been thoroughly discussed by Crampton,[4] Newcum[2] and St. Onge.[3] The theoretical gain is about 0.5% per degree subcooling of warm liquid. Great care should be taken in their application to avoid suction-line pressure drops which can overbalance the possible gain.

The **gravity feed system** has long been employed to control refrigerant flow through evaporators, particularly those comprising some form of pipe coil, although it has also been applied to shell-and-tube type apparatus. Its main function is to insure thorough flooding of the evaporator, at the same time protecting against liquid carry-over to the compressor. It consists of a surge drum elevated above the evaporator, a liquid feed leg connecting the bottom of the surge to the bottom of the evaporator, and a return line from the top of evaporator to surge drum, as shown in Fig. 12. In principle, it is based on the fact that the return leg, including the evaporator, being a mixture of liquid and gas, is lighter in weight than the liquid feed leg containing solid liquid. The effect is not only to flood the evaporator, but actually to recirculate liquid refrigerant, the rate of recirculation being dependent on the difference in weight of the two columns, a function of the heat per sq ft of evaporator surface. The surge drum serves as a separating chamber to return liquid to the evaporator and insure dry gas to compressor. Liquid level in the surge drum is usually controlled by means of a low-pressure type float expansion valve, although drain-type floats and thermal valves are sometimes used.

Liquid and return lines and pipe coil circuits should be sized for very low friction, as the success of the system depends on circulation by means of gravity flow only.

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27. LOW TEMPERATURE SYSTEMS

I. Operation and Practice

PRIOR to World War II applications of refrigeration equipment for the production of temperatures below approximately −40 F were unusual, except for cases of liquefaction of air and various other gases. Due, however, to aircraft developments and the higher altitudes at which planes were flying, it became necessary to develop a program for "altitude testing" of the various components of an aircraft system that were affected by low temperatures and high altitudes. These developments added to the need for test facilities at temperatures ranging from −40 F on down.

For refrigerant temperatures above −40 F single-stage operation is generally satisfactory. This, however, can only be achieved under favorable circumstances with normal condensing temperatures, due to the high compression ratio across the refrigerant compressor and the consequent inefficiency of the compressor. In general then, the production of temperatures below −40 F requires a somewhat special technique, different from that commonly used with the ordinary refrigeration compressor.

Freon Equipment Well Adapted to Low Temperatures

Ammonia, long used for refrigeration at all levels of temperature, has been successfully employed for the production of refrigeration temperatures as low as −90 F. This is approximately the low level for ordinary use, inasmuch as it approaches the triple point of −107.86 F. Very satisfactory results have been obtained using ammonia, but most manufacturers employ low-speed compressors for ammonia duty as compared with the commonly used high-speed Freon compressors. In addition, the compact, highly efficient and inexpensive low-sides that can be employed with the Freon refrigerants are not generally available for use with ammonia.

Freon-12 and **Freon-22** are also commonly used for the production of low refrigerant temperatures. Freon-12 is satisfactory for use down to approximately 0.5 psia, which corresponds to an evaporator temperature of −125 F. Freon-22 can be used to a somewhat lower temperature if the same limitation on absolute pressure at the compressor suction prevails, making the low limit for this refrigerant about −137 F. However, when using reciprocating compressors it is inadvisable to apply these limits, −115 F being a practical limit for Freon-12 and −125 F for Freon-22.

To illustrate the compression ratios involved in the compression of various refrigerants from low temperatures, **Fig. 1** shows the overall compression ratio for ammonia, Freon-12, and Freon-22 from suction temperature to a fixed condensing temperature of 100 F.

At the lower temperatures and extremely low absolute pressures, a very low density gas is available at the suction of the compressor. It is therefore essential that **suction valves** with little or no spring tension be employed to insure the filling of the cylinder in the suction stroke.

When handling rare gas, ample consid-

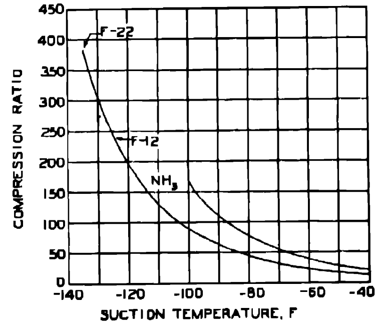

Fig. 1. Compression Ratio versus Suction Temperature at 100 F Condensing Temperature for Freon-22, Freon-12, and NH_3

eration must also be given to the effect of **suction line superheating** of the gas as well as **cylinder reheating** inside the compressor. Due allowance must therefore be made for the increase in gas volume which requires greater compressor displacement than would theoretically be considered. Centrifugal compressors and rotary compressors have also been used satisfactorily as boosters on low-temperature refrigeration, when the volume of gas handled warrants the selection of these somewhat larger compressors. Use of the centrifugal compressor and rotary compressor has been very satisfactory, due to lack of difficulty with the valves, as mentioned above with the reciprocating compressors, and also due to the minimizing of reheating of the gas when entering the compressor.

Compressor displacement may be obtained from **Fig. 2**, which shows cfm per ton for the various refrigerants. In determining these displacements, it is necessary to use the volumetric efficiency for the compressor involved. These displacements have been computed from the thermodynamic properties of the refrigerants, using a reheat factor based on figures obtained in practice. Furthermore, it has been assumed that a maximum compression ratio of 6 was used on the compressor, which in turn fixed the maximum amount of liquid cooling available.

Except for the fact that Freon-22 has higher absolute pressures and therefore can be used satisfactorily to somewhat lower temperatures, there is little to choose between Freon-12 and Freon-22. Freon-22 requires somewhat smaller compressor displacement, permitting a first cost saving in this item. The compressor, however, runs somewhat hotter since Freon-22 has a higher adiabatic exponent than Freon-12. Since the heat transfer equipment will be identical for Freon-12 and Freon-22 and the motor horsepower will be about the same, the overall saving in the system is small with Freon-22. Its use, however, is justified in many cases on the basis of the small compressor size and somewhat higher suction pressures involved. A further practical advantage of Freon-22 is that smaller refrigerant quantities are in circulation through the mains which, coupled with the higher density, leads to somewhat smaller line sizes, with a consequent saving in cost of the line and insulation.

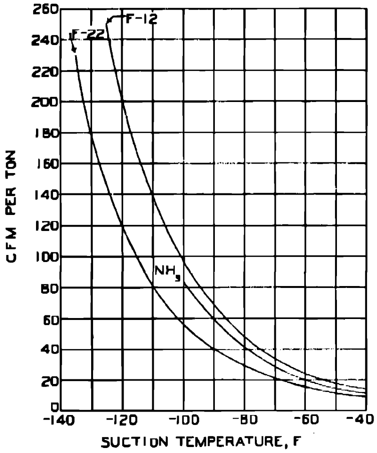

Fig. 2. CFM/Ton versus Suction Temperature at Liquid Temperature Based on 6 Ratios of Compression

As previously mentioned, Freon equipment is preferred to ammonia equipment due to the high speeds used with Freon equipment and the resultant saving in size involved. It should also be understood that Freon is preferable because of the inherent advantage of a refrigerant that is nonflammable and non-toxic. Its hazard in use is considerably less than that of ammonia.

Multi-Stage Compression Systems Required

Whether a two or three-stage compression system is to be used is dictated by the limitations of compression ratio on the compressors involved, as well as by operating cost and first cost. This is true when deciding whether to run one or two-stage compression, or whether to run two or three-stage compression. In general, practice has indicated that, for temperatures below −75 F, three stages of compression are usually required.

In order to give some idea of the horsepower per ton requirements, **Fig. 3** has been prepared, showing the approximate indicated horsepower per ton for the various refrigerants employing two or three-stage compression.

For the production of temperatures below −40 F multi-stage compression in

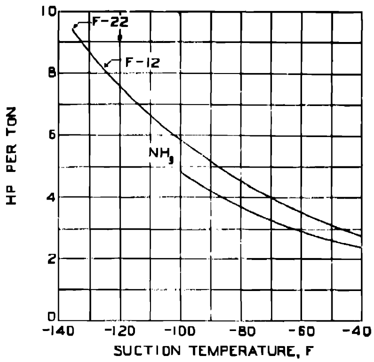

Fig. 3. Indicated Horsepower Per Ton versus Suction Temperature at 100 F Condensing Temperature

some form is generally required. The first and most simple method of multi-stage compression involves the use of **compressors in series,** the discharge of the low-stage compressor being the suction of the intermediate or high-stage compressor. Usual practice limits the selection of compressors to two or three stages.

The second method, called the **cascade system** of refrigeration, employs a series of gases of progressively lower boiling points. Each gas is used as a cooling medium to condense under pressure the gas of the next lower boiling point. This method is often used in the liquefaction of nitrogen.

For the common applications of low-temperature refrigeration encountered during the war, there is little justification for using a cascade system in the range of temperatures from −40 F down to approximately −135 F. Below the figure of about −135 F its use is justified with a refrigerant such as ethylene or several refrigerants of the Freon series which are not yet readily available. In these cases the cascade system is justified only because the pressures involved at normal condensing temperatures are so high as to require special high-stage compressors. Advances in the art of low-temperature refrigeration have indicated that there is little or no reason for cascading when using Freon-12 or Freon-22 as the low temperature refrigerant.

One of the objections to the use of a cascade system lies in the fact that there must be an overlap of temperatures in the condenser for one refrigerant, which condenser is the evaporator for the next higher stage refrigerant. This overlap of temperatures often leads to inefficiencies in operation, with somewhat greater displacement and horsepower being required. Furthermore, it has been found to be extremely difficult to operate a cascade condenser with a pressure much below 5 psig. Attempts to operate in a vacuum or at approximately atmospheric pressure have not been too successsful. due to the fact that a small amount of air in the condenser cuts the condenser performance considerably. A further difficulty with cascade condensers when operating at low condensing pressure is that the differential between the evaporator pressure and condenser pressure is so small as to make difficult the use of commonly employed refrigerant control devices, such as thermal valves or floats. The low pressures involved makes the use of extremely large control valves mandatory.

Use of Intercoolers

It therefore appears that compound compression or multi-stage compression is desirable whenever conditions permit. The operation of a multi-stage compression system requires cooling of the gas between stages in order to prevent overheating of the compressor when taking in a highly super-heated gas with a consequent rise in temperature through the compressor. At the same time it is desirable to employ **liquid cooling** between stages to increase the availability of the liquid delivered to the evaporator and thereby reduce the volume requirements of the low-stage compressor. **Intercooling** is more important with ammonia equipment than with Freon equipment because of the higher adiabatic exponent for ammonia. Liquid cooling is, however, more important with Freon than with ammonia, since the Freons have lower latent heats and any increase in availability is therefore of more significance.

With multi-stage compression, and with the discharge temperatures generally encountered, it is essential to provide for inter-stage gas cooling. With Freon refrigerants this is readily accomplished with small equipment and thermal valves. Unfortunately, thermal valve control of ammonia intercoolers is not feasible, making the use of float-controlled intercooling equipment necessary.

Intercooling equipment has two duties to perform: (1) to cool the gas to each machine; and (2) to cool the liquid ammonia on its way from the high-stage condenser to the low-stage evaporator. The most important function is that of gas cooling, to avoid burning up the machines due to high discharge temperatures. The least important function is the liquid cooling, since the ammonia liquid cooling is not so important as with Freon due to the difference in latent heat of the two refrigerants.

Two types of **ammonia-cooled intercoolers** are generally used for a combination of gas and liquid cooling:

1. Flash-type gas and liquid cooler
2. Shell-and-coil gas and liquid cooler.

Both types cool gas by bubbling it up through a body of liquid at some intermediate temperature. The **shell-and-coil cooler** provides cold liquid by means of a surface-type heat exchanger, in which a pipe coil is submerged in cold liquid of intermediate temperature, and through which coil the liquid to be cooled flows on its way to the low-stage evaporator. The **flash-type cooler** separates the liquid and flash gas resulting from the expansion of liquid at condenser pressure to the intermediate pressure; in the course of the expansion, the liquid is cooled to intermediate temperature.

Figs. 4 and 5 show diagrammatically these two forms of intercoolers for use with ammonia. They are generally sized as a gas and liquid separator, insuring a velocity sufficiently low that liquid may be separated from the gas which is withdrawn from the shell on its way to the compressor.

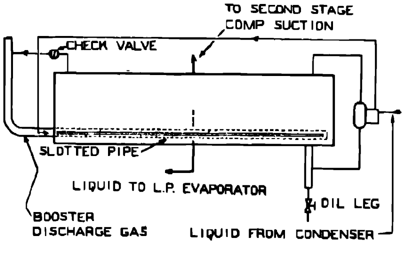

Fig. 4. Flash-Type Intercooler

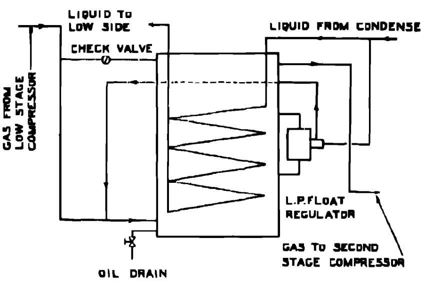

Fig. 5. Shell-and-Coil Gas and Liquid Cooler

Both types of ammonia intercoolers have merit. Utilizing the flash-type cooler makes it possible to cool the liquid proceeding to the evaporator down to the temperature corresponding to the intermediate pressure, at a sacrifice, however, of pressure differential which can be used for the operation of liquid control valves at the evaporator. The advantages are two-fold in that a slight thermodynamic gain is obtained with the lower liquid temperature, and a simple piece of equipment is possible, since a bare shell is used in place of a surface-type cooler.

The flash-type cooler, however, adds several disadvantages. The first is the loss of pressure differential, which means large liquid control valves and possible sluggish action. Another disadvantage is that any heat pick-up in a long liquid line between the flash-type cooler and the liquid control valve means that gas is generated in the liquid line. Such gas is extremely difficult to get through the liquid control valve and as a consequence the control valve becomes gas-bound. On three-stage ammonia systems the last stage of liquid cooling should always be made shell-and-coil type; if this is not done, on this third stage there would be very little pressure differential remaining to drive the liquid into the evaporator.

When only gas cooling is required between machines, the flash-type cooler may be utilized with the omission of the liquid connection to the LP evaporator.

A piece of intercooling equipment often used in connection with ammonia systems

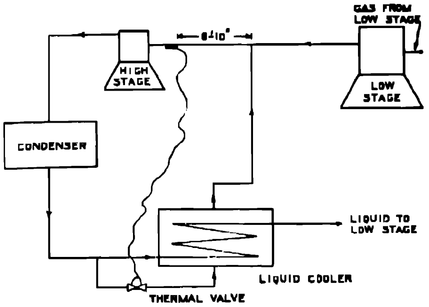

Fig. 6. Freon Liquid and Gas Cooler Hook-up

is a **water-cooled gas cooler.** Due to the fact that the discharge temperature from some of the compressors is sufficiently high to permit cooling by the use of water, these coolers are sometimes used. The amount of heat that they take out of the gas is, however, so small that no appreciable operating saving can be made. The slight savings that can be made in operating cost are in most cases overshadowed by the initial investment requirement for the piece of equipment as well as the additional cost of installation.

Both flash-type and shell-and-coil type liquid and gas coolers were used at one time on Freon systems, but they have been largely replaced by the shell-and-coil type liquid cooler, which is considerably simpler and is used only for cooling of the liquid. Gas cooling is obtained by proper mixing of the wet vapor with superheated discharge gas from the compressor. **Fig. 6** indicates a typical hook-up showing the position at which the thermal valve bulb is to be located to provide for the proper gas cooling.

Use of Oil Separators

The oil problem in connection with the compressors used on low-temperature refrigeration systems has been minimized by the use of oil separators. On a multi-stage compression system using shell-and-coil intercoolers, the **discharge line oil separator** on the high-stage compressor is all that is required to keep the oil in circulation in the system to a minimum. On systems using flash-type intercoolers it is essential to use a discharge line oil separator not only on the high-stage compressor but also on the compressor discharging into the flash-type intercooler. The use of discharge line oil separators keeps the oil where it belongs —in the compressors, allowing a minimum to go over to the low side.

In the case of ammonia systems, oil on the low side is detrimental to heat transfer effect, since the oil film has low thermal conductivity and tends to produce poor heat transfer. This same effect is not, however, encountered with Freon systems, because in the case of Freon-12 the oil and Freon are completely miscible, and it takes considerable oil concentration in the refrigerant to effect heat transfer. In the case of Freon-22, there is a separation of oil and Freon and the oil forms a separate phase lying on top of the Freon, not blocking the heat transfer surface nor causing too much difficulty in operation.

Oil can, however, cause some difficulty in ammonia and Freon-22 systems in connection with the operation of the liquid control devices. If too much oil circulates with the liquid refrigerant, it tends to accumulate in the liquid control valve, gumming up there and blocking off the flow of refrigerant through the valve. The use of the discharge line oil separator generally keeps down the flow of oil with the liquid to the point where no difficulty is encountered.

Due to the low mass flow of refrigerant through the compressor when operating at these low temperatures, the compressors tend to overheat. For small-size compressors with splash lubrication the surface area of the compressor is generally sufficiently large to dissipate this heat without difficulty. However, with larger compressors this is not the case, but fortunately many of these compressors utilize forced feed lubrication of the various rubbing surfaces, and, therefore, it is possible to add an external oil cooler to the oil circuit. This effectively removes the heat and maintains the oil at a temperature that does not cause difficulty in operation.

Sizing of Motors

As in all low temperature applications, the compressor can run under heavy loads during periods of high suction pressure, especially when starting when the system is warm. To take care of this situation it is therefore necessary to make adjustments to the motor horsepower and thereby prevent serious overloading of the motor during the starting period. Careful consideration must be given to sizing the motor, or special attention must be paid to the operation of the system when starting initially. If the compression system is to operate only at a fixed low temperature, it is generally possible to avoid the oversizing of motors to any great extent, providing for careful operation when the system is first put on the line. This method will permit putting in motors for the actual operating brake horsepower without any additional motor sizing.

For those cases where a definite pull-down time is required for the reduction in temperature from ambient conditions to some extremely low temperatures, the compression system will be operated at high suction pressures for considerable periods of time. For this reason it is necessary to oversize motors. General practice is to drive the high-stage compressor with a motor that will carry the compressor at the highest expected evaporator temperature. This is generally the "air conditioning" rating of the unit, in the case of Freon units. For intermediate or low-stage compressors, it is generally sufficient to size the motor to take care of double the balance load indicated horsepower plus the friction horsepower.

In line with recommending larger motors in cases requiring a pull-down, it is also necessary to give some consideration to the condenser that is used. It is generally advisable to oversize the condenser to some extent, certainly making it larger than the size required for the balanced load condition. Water-cooled condensers are not too critical in this regard, but extreme caution must be taken with economizer or evaporative condenser applications, since this type of condenser is more easily overloaded than the water-cooled condenser.

Use of Low Sides

Some mention should be made of the low sides that are generally used in low temperature refrigeration. With ammonia systems, **gravity-feed air coils** are frequently used with a large surge drum, refrigerant flow being controlled by a high-side float. Flooded **shell-and-tube coolers** are also used for cooling liquids such as trichlorethylene or methylene chloride.

For Freon-12 or Freon-22, various types of coils are used for cooling air, depending to a great extent upon the temperature level and the type of application. For rooms having considerable mass in which a slight variation of temperature of the air off the coil is not critical, thermal valves and direct expansion coils can be used satisfactorily for air temperatures down as low as approximately −70 F. Over the same temperature level, for wind tunnel applications, the direct expansion coil does not give as satisfactory results because of fluctuation in air temperature during a thermal valve cycle. For this reason gravity-feed coils have been utilized to avoid the cycle of the thermal valve and maintain reasonably uniform air temperatures leaving the coils.

As temperatures get well below −70 F, it is not possible to allow the necessary refrigerant pressure drop through the direct expansion coils, nor is it always possible to take the static head penalty that results with the gravity-feed coil. It is therefore essential on these applications to resort to pump recirculation of the refrigerant through the coils, using a down feed of the evaporator coil and thereby preventing static head penalties. In this type of pump recirculation, evaporation of the refrigerant takes place inside the tubes of the evaporator and a mixture of gas and liquid returns to a pump receiver in which the liquid and gas are separated, the liquid recirculating by means of the pump and the gas returning to the compressor.

Another type of liquid recirculation is sometimes used on temperatures below −70 F, depending to a great extent upon the size of the job and the space requirements for the evaporator itself. This method involves the **circulation** of the **refrigerant** in the **liquid phase,** refrigeration effect being taken care of by a warming up of the liquid refrigerant as it passes through the evaporator. The liquid refrigerant which is maintained under pressure is then allowed to flash into a large vessel, this flashing serving to reduce the liquid refrigerant to the pressure and temperature being maintained by the compressor. The liquid and flash gas are separated in the vessel and the liquid is then recirculated by means of the pump through the coil. The gas is taken to the compressor to complete its cycle.

For cooling liquids such as trichlorethylene or methylene chloride, shell-and-tube equipment is used with flooded operation for the small sizes where static head penalty is not too severe, and for larger sizes the liquid refrigerant is sprayed on the outside of the tubes.

Selection and Operation of Controls

Numerous types of liquid refrigerant control are available for use on low temperature systems. For both ammonia and Freon, **high-side floats** are preferred to low-side floats if there is any choice. High-side floats operate at a much higher temperature level as far as the float body and orifice are concerned, and therefore are less susceptible to difficulties from temperature variations or oil in circulation.

Where liquid coolers are used for precooling of the liquid on the way to the evaporator, a pilot-type high-side float is necessary. This is true inasmuch as it is desirable to keep the condenser drained, but at the same time insure a seal on the liquid line to evaporator.

Fig. 7 illustrates a form of pilot-type float which uses a float switch in conjunction with a solenoid valve close to the evaporator. Low-side floats can also be used where there are multiple evaporators. Care must, however, be taken to avoid congealing oil in the float body which can cause considerable difficulty. Liquid legs from the evaporator to the float must be carefully trapped and the line sizes should be kept large.

It should be noted that neither type float automatically provides for oil return to the compressor on Freon systems. It is sometimes possible to utilize an **oil still** on a float control job. The oil still is best used in conjunction with a liquid refrigerant pump, which can be used to provide sufficient lift for a small sample of the refrigerant that can be put through the oil still. The oil still merely takes heat from the liquid refrigerant at a higher temperature, boiling off the refrigerant and sending a mixture of refrigerant gas and liquid oil back to the compressor.

Automatic oil return can be obtained on Freon systems using a properly installed **thermal valve feed.** Thermal valves can be used for direct-expansion coils or can be used to feed refrigerant to a surge drum on gravity-feed jobs. In both cases it is advisable to utilize a superheater in conjunction with the evaporator, to provide dry refrigerant gas returning to the compressor and also give sufficient power for the operation of the thermal valve.

The operation of the system also calls for the use of a suitable low temperature lubricant. This is not necessary from the standpoint of the compressor, but once it is recognized that some of the oil from the compressor inevitably passes over to the low side, its need can be seen. The oil that is used should be water-free for use in all refrigerant systems and should also be as wax-free as possible. It has generally been noted that wax comes out of solution at lower temperatures and can easily accumulate at control valve orifices, thereby causing a restriction. The oil should also have a sufficiently low pour point that it does not congeal in the lines. This is more important with ammonia than it is with Freon-12, since the solution of Freon-12 in the oil effectively depresses the pour point. This is true also to some extent in the case of Freon-22, although separation of oil and Freon into two separate phases takes place at temperatures as high as 20 F.

The control of the refrigeration of low temperature systems depends to a considerable degree upon the application itself.

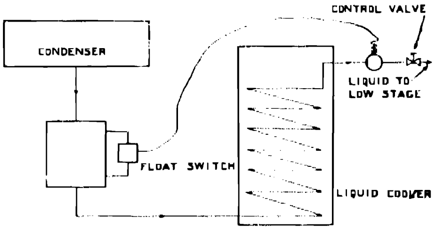

Fig. 7. Pilot High-Pressure Float

Starting and stopping of the compressors has been utilized to maintain temperatures in large spaces where there is considerable flywheel effect. It is necessary, however, to have some temperature fluctuation in order to actuate the temperature controller, which causes the temperatures to vary inside the conditioned space. Where these variations are not too serious, and sufficient temperature change can be utilized to prevent short cycling of the equipment, reasonably satisfactory results can be obtained. For multi-stage compression systems this means that some provision must be made for time delay between motor starters in order to prevent all machines from coming on the line at once.

If too close a temperature control is not required, it is often possible to permit the machines to rebalance at lower temperatures, since as suction temperature is reduced machine capacity falls off very rapidly. This slip in temperature is generally not too great as long as excess capacity has not been provided for a pulldown or for large operating loads that may fail to exist.

It is also possible to throttle the suction to the low-stage compressor, thereby providing for temperature control. This is generally dangerous from the standpoint of the machine since it causes the machine to run considerably hotter than normal, and, in the case of forced feed lubrication of the compressor, may cause the oil pump to fail because of too low an absolute pressure suction to the oil pump. When the dangers involved in the use of a throttled suction are recognized and suitable provision is made against the difficulties that may be encountered, this method does give reasonably good control.

One of the best methods for temperature control that can be made automatic is "hot gas bypass," which provides for artificially maintaining the load on the compression system by supplying gas from the high side through a control valve, supplementing the gas that is obtained from the low side. As the load falls off and less gas is provided by the low side, more gas is provided through the hot gas bypass control valve. This method takes more operating horsepower but gives very close control of the system.

II. Gas Liquefaction and Dry Ice

1. Low temperature refrigeration systems had their **laboratory development** in the liquefaction of the so-called "permanent gases" such as air, oxygen, nitrogen, and later hydrogen and helium. The first commercial development of low temperature refrigeration was effected by the attempt to extract pure oxygen from air by first liquefying air and then recovering the oxygen in a relatively pure state by fractional distillation. After the principle of separating a gas mixture by liquefaction and distillation was successfully applied to air, the technique was extended to the recovery of helium from natural gas, and hydrogen and ethylene from petroleum cracking still gases, as well as to the separation of the rare gases such as argon, xenon, and krypton from air.

In some of the simple types of gas liquefiers the gas itself acts as the refrigerant. In other types, greater efficiency is obtained by employing separate refrigerants. In all liquefying machines the gas must be under sufficient pressure that it can be allowed to expand to a lower pressure, causing a lowering in temperature by either of the two following methods:

a. Joule-Thomson expansion—adiabatic irreversible expansion of gas at high pressure to a lower pressure with a lowering in temperature. At the end of the expansion process the gas is at a lower temperature and lower pressure, with an entropy increase.
b. Expansion of the gas reversibly or irreversibly in a machine in which it can perform work; the gas undergoes a decrease in temperature, pressure, and enthalpy.

For a particular gas at a chosen initial temperature and pressure expanding to a chosen final pressure, the expansion (with work produced) will result in a greater lowering of temperature than the adiabatic Joule-Thomson expansion. However, only in recent years has the expansion principle been used extensively.

In all liquefying machines the gas cannot expand from a higher pressure to a lower pressure and have a sufficient reduction in temperature for liquefaction to occur, but the gas must first be cooled to a sufficiently low temperature that the final expansion is below the critical temperature. For the cooling of the gas before expansion another essential part of the liquefying machine, aside from the compressor and expansion valve, is a heat interchanger. Combination and modification of these three essential parts are used in various ways, as further described, in refrigeration systems.

Linde or Hampson System

2. This is the simplest type of liquefaction system, consisting of the three minimum parts, i.e., compressor, heat interchanger and expansion valve. **Fig. 8** is a schematic flow diagram of the liquefier.

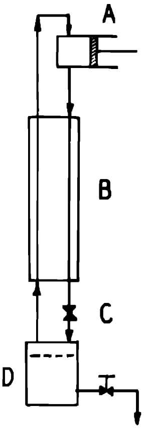

Fig. 8. Linde System

The air leaving the compressor A is under a pressure of 100 to 200 atmospheres, cooled to about −120 C after leaving the heat interchanger B, and expanded adiabatically to atmospheric pressure with a further drop in temperature to the boiling point. If the gas is compressed to 200 atmospheres, about 10% of the gas leaving the expansion valve C liquefies; the remaining unliquefied air passes through the cold side of the heat interchanger, cooling the high pressure gas from the compressor, after leaving the interchanger at a temperature about 2 or 3 degrees above room temperature. The quantity of gas that liquefies after an expansion is a function of pressure and the thermal properties of the gas.

This type of liquefier, because of its simple construction, is often used where small quantities of liquefied gas are needed. However, if a more efficient machine is needed, this system can be modified and is designated as the improved Linde system.

Improved Linde System

3. In order to improve the efficiency of the Linde liquefiers a two-stage expansion and compression system was devised. **Fig. 9** indicates the arrangement of this system. The air is compressed by the first-stage compressor A to 40 atmospheres; the second-stage compressor B further compresses the gas to 200 atmospheres. The compressed gas then passes through the inter-coolers C and expansion valve D, where the first adiabatic expansion reduces the pressure to 40 atmospheres into chamber E. Here 20% of the gas is further expanded to 1 atmosphere as it passes through expansion valve F into chamber G. For the final expansion about 8% of the original gas is liquefied.

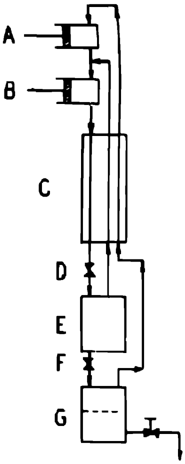

Fig. 9. Two-Stage Linde System

No liquid will result from the first expansion unless the temperature in chamber E is below the critical pressure. For air this would have to be below −222 F. The optimum pressure for minimum power consumption per unit quantity of liquefied air produced is the same as the pressure given above, namely 40 atmospheres.

From chamber E about 20% of the air under 40 atmospheres is expanded on further to chamber G, while the remaining 80% is returned to the compressor for recompression.

The expanded air from chamber F and part from chamber D pass counter-current to the incoming compressed air in intercooler C and then on to their respective compressor intakes.

Economy in power expenditure is achieved in this scheme because a major portion of the air (about 80%) is compressed from 40 to 200 atmospheres instead of from 1 to 200 atmospheres. Since the energy required to compress a gas varies as the logarithm of the ratio of the pressures, the work of compression from 40 to 200 atmospheres is about $\frac{1}{8}$ the work of compression from 1 to 200 atmospheres. Further power economy can be achieved by precooling the compressed air by an auxiliary refrigeration cycle.

Claude System

4. As previously mentioned, the expansion of a gas under reversible conditions produces a cooling effect which is greater than an adiabatic irreversible expansion from the same initial condition to the same final pressure. The application of the above stated principle alone in the expansion of a gas in a cylinder to liquid has been found to be impractical at low temperatures, because of the low efficiency of expansion and difficulty of lubricating the cylinder.

The Claude system is a combination of both principles of cooling, i.e., the Joule-Thomson effect and the approximate reversible expansion of the gas in an expansion engine.

The Claude liquefier is schematically given in **Fig. 10.** The air is compressed to about 40 atmospheres by compressor A; the compressed air passes through the interchanger B and separates into two parts. One stream, about 20%, passes through the interchangers C and D and is finally expanded in valve F to container E at atmospheric pressure, where the liquid and vapor portions separate. The unliquefied part from the expansion returns to the intake of the compressor after passing through the three intercoolers D, C, and B. The remaining 80% of the stream that separates from the main stream between interchangers B and C is expanded in cylinder G of the expansion engine, and joins the returning low-pressure air from the interchanger D on its way back to the intake of the compressor.

The theoretical power consumption for a given quantity of liquid produced is a function of the exhaust pressure of the compressor A, the fraction of the air that passes through the expansion valve and the inlet temperature of the air to the expansion engine. The conditions given above with an inlet temperature of about −80 C are optimum for power consumption.

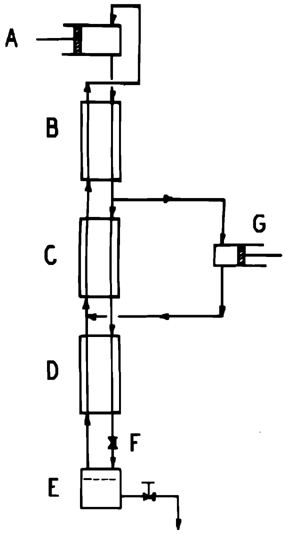

Fig. 10. Claude Liquefier

The difficulty of lubricating the cylinder of the expansion engine can be overcome to some extent by using leather or fiber piston rings, and according to Russian practice[6] the engines have been successfully replaced by turbines in which the lubricating problem at low temperatures does not arise as the turbine bearings are at relatively warm temperatures at which lubricating oil can be used.

The power generated by the expansion engine is sometimes used to "help" the compressor through a belt connection. In other cases, the expansion engine may run a dynamo.

Heyland System

5. If the compressor discharge on the Claude system is raised to 200 atmospheres the optimum inlet temperature for the expansion engine will be at approximately room temperature and the first intercooler, as shown in Fig. 10, is not needed. The conditions just described are requisite to the Heyland system. Such conditions are of a practical advantage in that the difficulties of lubricating the cylinder of the expansion engine are lessened. Light lubricants have been found satisfactory.

In the Heyland system minimum power

consumption for the 200 atmosphere compressor discharge is achieved by passing 55% of the air through the expansion engine and the remaining 45% through the valve. Thus this liquefier differs from the Claude liquefier in which 80% of the air is expanded in the engine.

Cascade System

6. The cascade system of refrigeration employs a series of gases of progressive lower boiling points. Each gas is used as a cooling medium to condense under pressure the gas of the next lower boiling point. This system was first proposed and employed by Pictet[3] in 1877 to liquefy oxygen. The intermediate gases chosen were sulfur dioxide and carbon dioxide. Kammerlingh-Onnes[4] developed a cascade system to liquefy oxygen and nitrogen, using as intermediate gases methyl chloride and ethylene.

In the cascade system proposed by Keesom[5] for liquefying nitrogen the following gases were employed in this respective order: ammonia, ethylene, methane, and nitrogen. This system is indicated in **Fig. 11.** The ammonia is compressed to 10 atmospheres by compressor A_1, liquefied at room temperature and expanded in valve D_1 to 1 atmosphere. The ammonia that vaporizes from evaporator C_1 passes through the interchanger B_1, cooling the compressed ethylene from compressor A_2. The ethylene under a pressure of 19 atmospheres is liquefied in the submerged coil of evaporator C_1. The liquefied ethylene expands in valve D_2 to atmospheric in evaporator C_2. The vaporized ethylene from C_2 passes through the interchanger B_2, cooling the methane from compressor A_3.

The methane is liquefied under a pressure of 25 atmospheres in the submerged coil of evaporator C_2, expanded in valve D_3 to 1 atmosphere in evaporator C_3. The vaporized methane from the evaporator C_3 is divided into two streams, passing through the interchangers B_2 and B_3, cooling the compressed methane and compressed nitrogen streams respectively. The nitrogen at a pressure of 19 atmospheres from compressor A_4 liquefies in the submerged coil of evaporator C_3. The liquefied nitrogen is expanded in valve D_4 to 1 atmosphere. The liquefied portion from the expansion may be withdrawn from accumulator E while the vaporized portion returns to the compressor after first passing through B_3.

The cascade system of obtaining low temperature is unique in that the power consumption per unit of gas liquefied is less than that of the other processes described.

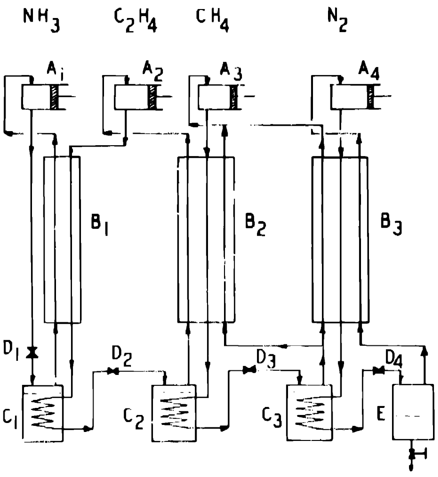

Fig. 11. Liquefaction of Nitrogen

7. In order to make a **comparison of the efficiency** of the various processes Keesom[5] made a detailed analysis of the power requirements for the production of 1 kilogram of liquefied nitrogen under 1 atmosphere, starting with gaseous nitrogen at 25 C and 1 atmosphere. **Table 1** is a tabulation of the result of this analysis on the power requirements for the cascade, Linde, and Claude processes. The theoretical power requirement for a hypothetical process operating under perfect reversible conditions is also listed in Table 1.

The reversible process requires the minimum amount of work and is independent of any mechanism. Actual processes have greater power requirements because there are energy losses inherent in any practical process as distinguished from a reversible process. The reversible process cannot actually be achieved, but it is set up as a standard for purposes of comparison.

In Keesom's analysis of the three processes the nature and amount of the energy losses are given. Aside from energy loss due to friction in the moving parts of the compressor and the fluid flowing through the system, there is the irreversible effect of heat transfer in the interchangers, evaporators, and heat leakage into the apparatus from the surroundings. The greatest losses occur in the expansion of the gas or liquid in the expansion valves. The nearer the compressed gas is to saturation conditions the less irreversible the expansion (indicated as less increase in entropy). Moreover, the expansion of the liquid in the valve has less irreversible effect than the gas.

Table 1. Theoretical Power Required, hp-hr per kg of Liquid Nitrogen

Hypothetical reversible process	0.29
Cascade process	0.73
Linde process	1.41
Claude process	1.19

The cascade system is more economical in power requirement because only liquids are expanded. If the compressed gas in the other systems is precooled with an external refrigeration system there will be an improvement in the overall power economy.

Commercial Liquefaction of Natural Gas

8. A recent commercial application of the cascade system was the liquefaction of natural gas for storage purposes. A plant erected at Cleveland, Ohio, was built for the purpose of liquefying the natural gas at times when the consumption was low, storing and re-gassifying it at peak load conditions when the pipe line from the gas field would not be able to furnish the necessary quantity.

The liquefaction of natural gas reduces the volume of required storage space 600 times. This plant was designed to liquefy 4,000,000 cu ft of natural gas per day and had a storage capacity of three spherical tanks each 57 ft in diameter. The liquefied natural gas was stored at a temperature of −250 F and at a pressure of approximately 12 psig. The storage spheres were made of nickel steel rested in cork insulation inside of a steel sphere 63 ft in diameter. From experimental work the heat absorption was estimated to be relatively small, and if the storage spheres were full, at the end of a year's time only half of the liquid would be evaporated.

Before liquefaction, the natural gas is processed to remove CO_2 and water vapor in order to prevent plugging of the lines, and nitrogen is vented off at intervals to prevent too great an accumulation in the

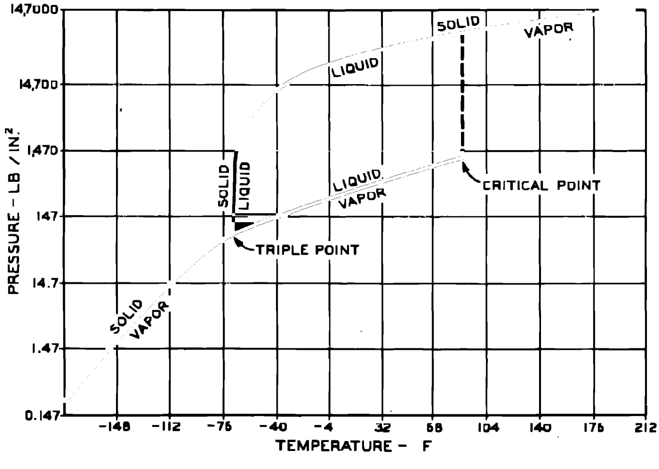

Fig. 12. Pressure Temperature of CO_2

recycle system. This plant was unfortunately destroyed by an explosion and fire.

Detailed descriptions of this plant have appeared in various publications.[1,2,3,7] In general, the refrigeration cycles used are the same as proposed by Keesom, with the exception of the nitrogen cycle, namely, ammonia, ethylene, and methane.

Dry Ice

9. **Carbon dioxide ice** or dry ice is useful as a refrigerant agent in many applications, especially in transportation. Its high cost of manufacture limits its field of use, but it has extraordinary merit in being readily handled and cut dry—sublimating to gas, non-corrosive and non-toxic, high as to latent heat and low in temperature. In the gaseous form it is a good insulator.

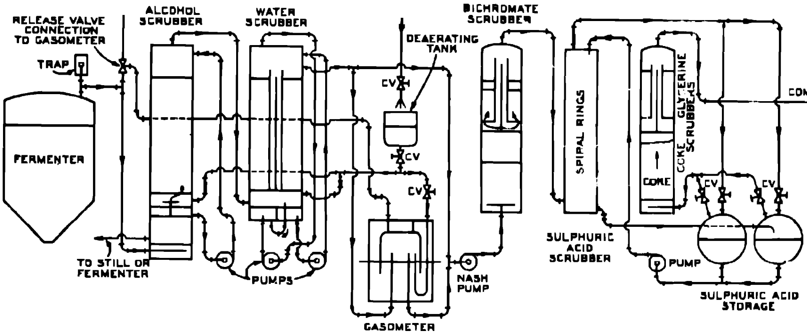

Fig. 13. Diagram of Reich Process of Carbon Dioxide Gas Production, Showing Purifier Apparatus

Manufacture of dry ice divides into the process of making or recovering the gas, a chemical process, and its liquefaction and solidification by a compression cycle.

Carbon dioxide has been obtained commercially from natural springs and fissures, from wells, from heating or acid treating the carbonates of alkalies and alkali earths, from fermentation of organic substances, and from the burning of carbonaceous substances.

The refrigerating effect is the sum of the heat of sublimation (latent heat of the solid) or 248, plus the sensible heat derived by raising the temperature to say 32 F, or about 27, giving 275 Btu per lb. See p. 129 for table of physical properties. Pressure-temperature relations are shown in **Fig. 12** and Chart 4, Chap. 7.

10. **Limestone, magnesite and dolomite** have all been used and are being used as sources of CO_2 gas, and appear to be the most rapidly growing sources. Two methods are used to separate the CO_2 from the carbonates: (a) by heat, (b) by acid treatment. In the first method, the stone is heated in kilns, preferably of the vertical type. The offcoming gases (containing the CO_2 from the carbonate and that from the combustion of the fuel used to heat the carbonate) are found to be from 25 to 45% CO_2. The yield of CO_2 is approximately 50% of the weight of stone used, e.g., the "ammonia-soda" process.

In the acid method the carbonate is treated with acid and the offcoming gas is practically pure CO_2. The yield is from 44 to 52% the weight of stone, dependent on the kind of carbonate used. This latter method appears to be applicable only where the salts produced from the acid treatment are marketable.

All **organic matter in fermenting** generates CO_2. Commercially the organic matter used is molasses and grain. The dry gas is about 99.5% CO_2 and can be recovered in proportions of about 5 lb of CO_2 per gal of alcohol produced. The odor of the impurities in this case makes satisfactory purification a difficult task. One of several methods is shown in **Fig. 13.**

Coke ranks next to limestone as a commercial source of carbon dioxide. Usually it is burned under boilers in such a manner as to obtain the maximum concentration of CO_2 in the flue gas. The steam generated is used for generating power and for aiding in the purification process. The coke should be of high carbon content, low in sulfur and of good structural strength; 72-hr foundry coke has been the standard of quality for years. Lately high grade by-product coke has been used satisfactorily.

Theoretically, 1 lb of coke (90%) will generate 3.0 lb of CO_2; actually, the recoverable yields have in the past been nearer a ratio of 1 to 1. Good practice with present equipment should yield a ratio of 2 lb of CO_2 per lb of coke burned. The flue gas, under good firing conditions, will run 15 to 18% CO_2, about 80% nitrogen and small amounts of oxygen, carbon monoxide, sulfur dioxide and hydrocarbon gases.

About four tons of CO_2 are generated in making one ton of steel. Most of it is generated in the **blast furnace** from the limestone and coke used. The gases as they leave the blast furnace itself run high in carbon monoxide, but on being burned in the stoves or in the blowing engines, the composition is about 20 to 30% CO_2, 70 to 80% nitrogen, with small amounts of carbon monoxide, oxygen and sulfur dioxide.

A vast potential source of CO_2 is found in the **petroleum industry** as a by-product in hydrogen production for hydrogenation, where CH_2 represents a waste hydrocarbon. The reaction may be written

$$CH_2 + H_2O \rightarrow CO + 2H_2O$$
$$CO + H_2O \rightarrow CO_2 + H_2$$

Fuel oil and natural gas have been suggested as sources of CO_2, but so far they have not proved commercially practical. They yield a flue gas of less than 15 and 10% CO_2 respectively, but with the other considerations such as low price of oil and gas and smoothness of operation, they may be attractive as sources of CO_2.

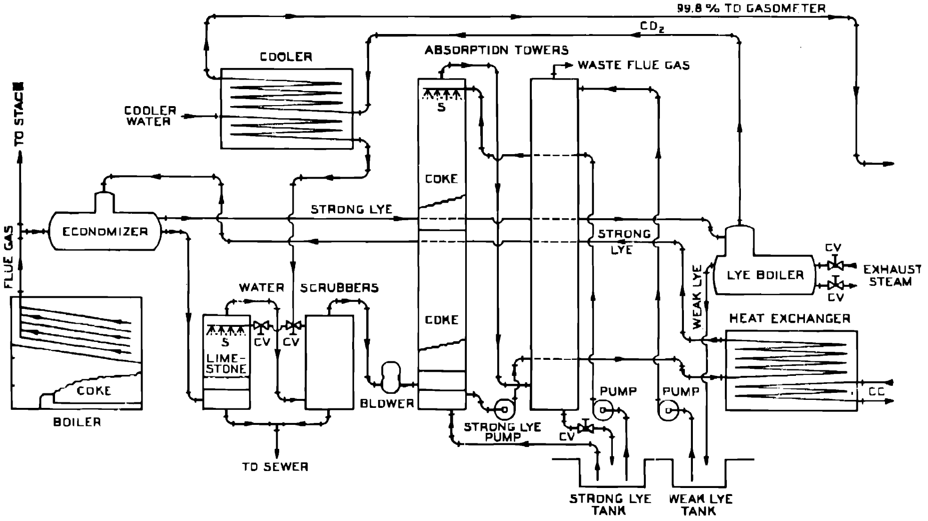

Fig. 14. Alkali Absorption Process

Underground sources of CO_2 have been found in drilling for oil or natural gas; several such sources have been profitably exploited in the U. S. in connection with dry ice. One source in Mexico is of commercial importance.

11. The **alkali absorption process** is applicable to those sources of CO_2 gas in which the other components of the gas mixture are relatively inert. It is used extensively in recovering CO_2 from the flue gases of coke and from limestone. It has been found commercially feasible for gases containing from 15 to 50% CO_2 and is the process most commonly adopted, **Fig. 14.**

The gases from the flue or from the kiln are first washed with water to free them from dust and sulfur dioxide and to cool them to about 110 F. This washing is done in steel or concrete towers, filled with limestone of about 6 in. diameter. These tanks are lined with acid-proof material to prevent corrosion. Water is circulated over these towers, and as it becomes charged with sulfur dioxide it is allowed to flow to waste.

The gases are next passed through the absorbers. In the type of plant most commonly built, these absorbers are steel towers, two in series, packed with coke, placed on grids at say 15-ft intervals in the height of the tower. Manhole plates are provided to permit replacing of coke when needed.

A solution containing from 7.5 to 8 lb of sodium carbonate or 12 to 12.5 lb of potassium carbonate per cu ft is circulated over the absorbers and, dissolving out some of the CO_2 in the flue gases, issues from the absorbers containing about 3 lb of sodium carbonate and 6 lb of sodium bicarbonate per cu ft.

The gas enters the absorbers with approximately 17% CO_2 content, and is vented to the atmosphere at about 8 to 9% CO_2.

From the absorbers the **strong lye** (mixed bicarbonate and carbonate solution) is pumped to a **lye boiler** where it is heated to about 240 F and thus freed of its CO_2 content. Chemically:

Weak lye:

$$Na_2CO_3+H_2O+CO_2=2NaHCO_3$$

Strong lye:

$$3NaHCO_3+\text{Heat}=Na_2CO_2+H_2O+CO_2$$

The CO_2 gas mixed with steam is cooled in order to free it from most of the water vapor and is sent to a gasometer or directly to the compressors.

The following operating conditions for plants using sodium carbonate will contribute greatly to proper results. Summer conditions have been chosen, because that is generally the time of peak load, when control is most important.

Temperatures (solution)	*F*
Leaving last absorber—entering interchanger	120–140
Leaving interchanger—entering lye boiler	220–230
Leaving lye boiler—entering interchanger	240–250
Leaving interchanger—entering absorbing system	140–160
At pump circulating solution over first absorber	135–155
Between first and second absorbers	130–150
At pump circulating solution over second absorber	125–145
Temperatures (gas)	
Entering scrubbers	300
Leaving scrubbers	130–150
Leaving lye boilers	210–235
Leaving gas coolers	70–100
Entering compressors	25– 60
HP liquid CO_2 leaving water coolers	50– 70
HP liquid CO_2 leaving refrigerating coils	20– 50
Analysis:	
Flue gas of CO_2 leaving furnace, %	15– 20
Flue gas of CO_2 leaving absorbers, %	6– 9
Solution leaving absorbers, sodium bicarbonate, lb per cu ft	5– 6
Solution leaving absorbers, sodium carbonate, lb per cu ft	3– 4
Solution leaving lye boilers, sodium bicarbonate, lb per cu ft	2 3
Solution leaving lye boilers, sodium carbonate, lb per cu ft	5– 6

A variation of this absorption system is used by the ammonia-soda industry, consisting essentially in absorbing the CO_2 from kiln gas in strong lye under slightly raised pressure and freeing the CO_2 in lye boilers, in which heat is applied and the pressure reduced. With the use of pressure, the size of equipment is reduced, and a considerable improvement in percentage of CO_2 gas recovered is claimed.

Another variation is the use of triethanolamine as an absorbent in place of an alkaline carbonate. The reaction is

$$\text{Triethanolamine: } 2N(C_2H_4OH)_3+H_2O+CO_2=\text{Triethanolamine carbonate: } (N(C_2H_3OH)_3H)_2CO_2$$

The triethanolamine carbonate breaks down on heating, releasing the CO_2. Theoretically, 6.77 lb of this compound can absorb 1 lb of CO_2.

Still another absorption process is one involving the solution of CO_2 in flue gas by means of water under relatively high pressures. After the water is carbonated to its maximum under the conditions, it is passed through a low-pressure vessel where the CO_2 is freed from its solution and passes on in a pure state. This method, while old, has met with very limited favor in this country.

Many plans have been drawn and several plants have been built in which CO_2 is freed by means of compression and fractional distillation. It is quite possible to freeze out the CO_2 by such means, but the pressures are excessive when applied to gases of low CO_2 concentration because of the partial pressure of the extraneous gases (nitrogen) present. High power costs and numerous operating and maintenance difficulties have caused the abandonment of the few plants that were so designed for handling flue gas. The process has possibilities for higher concentration sources.

12. **Removal of odors.** Another process using activated carbon has been used for purifying CO_2 from oil wells. The gas containing hydrocarbons and sulfur comes from the wells at about 1,000 psi, and is first passed through towers containing iron oxide to remove the sulfur compounds, and then through a series of towers containing activated carbon which absorbs the hydrocarbons. Revivification is accomplished by use of superheated steam at periods of every few hours. The oxide towers are revivified by passing air through the oxide at controlled rates of flow. The CO_2 gas, free from all odorous impurities, flows from the towers through cooling coils where, at the pressures involved, it liquefies and is ready to be solidified.

One of the processes peculiarly well suited for the removal of water-soluble odorous impurities is the **silica gel** process. In this process the gas is collected, usually from fermentation vats, and after a preliminary water scrubbing to remove the bulk of alcohol and other organic matter, it is passed through the first and second stages of compression—i.e., up to 380 psi, and then through the absorbers containing gel. When the absorber is saturated the gas is passed through a companion absorber while the first absorber is revivified with heated air.

A modification is the purification of low-pressure CO_2 with silica gel; the CO_2 from the fermenters is scrubbed and cooled to about 45 F and then at about 5 psi is forced through absorbers containing gel. The revivification is accomplished by hot air as above. This low-pressure system naturally requires larger equipment and is more costly. The manufacturers guarantee a product free from all odor.

Alumina gel is claimed to serve satisfactorily as a purifying medium for CO_2 gas.

Liquefying and Solidifying

13. The **compression** of CO_2 gas is usually accomplished in three stages, the first taking the gas from about atmospheric pressure up to 65 to 80 psi. The gas is then cooled and passes into the second stage of compression, where it is carried up to 300 to 400 psi, is again cooled, and often purified at this point. Then it is passed into the third stage where a pressure of 900 to 1,200 psi is reached. This CO_2 is usually liquefied in coils cooled with water. The liquefying temperatures for various pressures are found in the tables of thermal properties.

To convert this liquid into **solid** CO_2 the liquid may be (a) frozen by the removal of heat by external means, (b) evaporated so rapidly that a portion of it freezes, or (c) ejected through a small orifice or nozzle, under which condition a portion of the liquid freezes out as snow.

The first method was first proposed about 1895, and has been tried on a small scale but discarded. In the second method the liquid CO_2 is put in a pressure vessel surrounded by a source of cold. Inasmuch as the boiling point of CO_2 is a much higher temperature than its freezing point (−69 as against −109 F), such a source is not difficult to obtain. The liquid CO_2 occupies a much larger volume than the resultant ice, and for this reason it has been possible to supply the freezing chamber with liquid CO_2 under pressure during the freezing period. The ice produced is translucent and hard, and has a density of about 90 lb per cu ft.

When a cylinder of liquid CO_2 is opened with its valve uppermost and the gas is permitted to escape rapidly, a portion of the liquid is frozen into a porous disintegrated mass known by the descriptive name of "pop corn snow" by a constant enthalpy process as in the case of any expansion. The liquid may evaporate directly in a mold and the porous solid is pressed under hydraulic presses into dense ice. At the present time several commercial plants are utilizing modifications of this simple method.

The third general method of solidification is that involving the **expansion** of liquid CO_2 itself. When liquid CO_2 is forced through a nozzle the liquid flashes into gas and snow. The proportion of snow is dependent on several factors, particularly that of temperature of the liquid CO_2. Snow formed in this way is usually compacted by tamping and then pressed by hydraulic means. **Fig. 15** shows the power required per ton of dry ice for various cycles at various temperatures of the liquid.

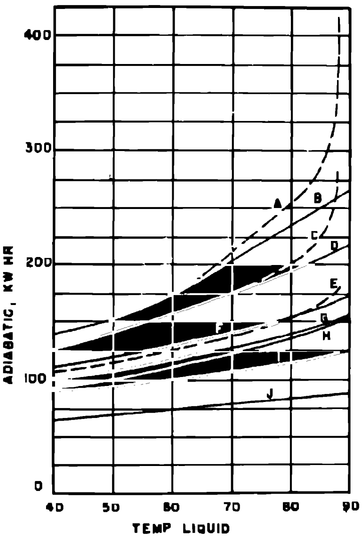

Fig. 15. Summary of Calculations of Theoretical Cycles of Snow Production

A, B, Simple cycle; C, D, same with precooler; E, F, bleeder cycle; G, H, bleeder with precooler; I, binary cycle; J-Carnot cycle. Dotted lines are based on saturation pressures, solid lines are based on optimum pressures.[1]

[1] Stickney, A. B., *Refrig. Eng.*, Dec. 1932.

Most of the various processes that are being used today are combinations of these three methods. Several of these make snow in a chamber maintained at about the triple point; in this way liquid and snow are injected at intervals into the chamber, forming a wet sludge. This is later compressed to form a dense ice. The Carba process injects the wet snow into the chamber with considerable force, and the machines are so designed that no further pressing is necessary to produce a dense ice.

Other variations in the simple conversion of liquid CO_2 into solid CO_2 arose in the methods advocated for the **precooling** of the liquid CO_2. Some prefer to cool the liquid by external means such as an ammonia refrigerating system, others by using the liquid CO_2 as a refrigerant, expanding it and recompressing it through the third stage of the compressors. Others expand a portion of the CO_2 between each stage of compression much along the lines laid down by customary refrigeration practice to obtain maximum efficiencies.

14. Storage and transportation. Because of the extremely low temperature of the solid, CO_2 does not lend itself readily to long-time storage. Two theoretically possible methods of slowing down the meltage are (a) mechanical refrigeration of the storage room or (b) by means of pressure applied to the storage space; neither is feasible because of the costs involved. The ice must serve as its own refrigerant.

Attempts have been made to store the CO_2 for several months in large insulated gas-tight silos. In these, ice made in winter and spring is to be stored for the summer peak, and the CO_2 gas liberated by the melting ice is returned to the compressors for reliquefaction and reconversion into ice. The impression is that these have not proved commercially advantageous because (a) it is difficult for men to work in the silos on account of the large proportion of CO_2 in the atmosphere, (b) the ice changes in crystal structure and becomes sugary, and (c) the capital expense and the labor costs are excessive. Storage for one month or six weeks has been successfully accomplished where the ice is of satisfactory crystal structure. These temporary storage boxes are so constructed that a man may load and unload and yet keep his head above the CO_2 gas level (see **Table 2**).

The **well-type** storage container is the only one commonly used today. The exterior walls and bottom are generally constructed of sheet metal and sealed against leakage of cold air. Several inches of effi-

Table 2. Dry Ice Losses per 24 hr, %

Shipping carton for 50 to 100 lb of dry ice, insulated with dry wood pulp	15–20
Shipping box for 200 lb of dry ice (3-in. insulation)	8
Shipping box for 2,000 lb of dry ice (5-in. insulation)	3
Shipping box for 4,800 lb of dry ice (6-in. insulation)	1.5
Insulated truck, well-type body	1.2
Insulated railway car, well-type cargo space	0.8
Plant storage, well-type storage box	0.3

cient insulating materials are interposed between outer and inner walls. A light hinged lid or detachable cover forms the top of container. A mattress is often provided to protect the top surface of the solid CO_2 stored within the box. Storage containers vary in size from 8 to 54 cubes capacity for standard 10 in. cubes.

Shipping boxes follow the design of storage containers except that weight is kept to a minimum through the use of less insulation and lighter structural members. Capacities of shipping containers for solid CO_2 vary from 1 to 8 cubes.

There are now in use specially designed trucks and cars for transportation of solid CO_2. These are of the top opening or well type with a number of separate compartments, each with independent covers.

Shipment of dry ice involves many difficulties, losses up to 30% being typical where the product is submitted to several handlings.

Dry ice from a chemical analysis is always of a purity above 99% CO_2, but for commercial use its quality is judged mainly for freedom from odor and for structure. Dry ice from fermentation sources is examined primarily for the odor of beer or molasses; from combustion of fuels, from lime kilns, etc., for odors of hydrogen sulfide or the odor of "smoke"; from natural wells for odors of hydrogen sulfide and gasoline; and from all sources for an odor of oil brought about by incorrect lubrication of the compressors.

In structure dry ice should be neither soft nor too brittle. Ice that is well made and is of satisfactory density may be sawed into thin slices 10 in. × 10 in. × $\frac{1}{4}$ in. and these slices be sufficiently rugged for handling. Densities of 85 to 95 lb per cu ft (50 to 55 lb per 10 in. cube) have been found most satisfactory.

Summary of Uses

15. As a refrigerant dry ice is convenient, economical in application to both stationary and mobile units, and susceptible to a readily controlled temperature range from the intense cold of frozen food to that of the moderate temperature of a butter box. It can profitably be used as a refrigerant in the transportation and dispensing of unfrozen perishable foodstuffs such as fish, meat, dairy products, fruits, and vegetables, as well as the frozen product. It is also used in the metal-working industry, electrical industry, chemical industry, in the laboratories, and in certain applications of food freezing and of air conditioning.

As a source of carbon dioxide gas, dry ice offers economies in cost of production, transportation and storage. It can be profitably used for the innumerable purposes to which carbon dioxide itself is put:

a. In carbonating beverages
b. For preserving perishables
c. In the canning and food packaging industry
d. In the chemical industry
e. In fire prevention
f. In the medical profession
g. As a source of power

Low Pressure Air Liquefaction

16. During the latter part of World War II and subsequently, extensive developments have been made for producing oxygen and nitrogen from air in low-pressure systems (less than 100 psi). This involves extensive use of efficient heat transfer surface and use of an expander engine.[9]

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28. AUXILIARIES AND THEIR OPERATION

BEFORE any refrigeration system is put into operation, it is necessary that certain precautions be taken in order to obtain good efficiency from the standpoint of both economical performance and maximum life of equipment.

In line with these precautions the most important considerations are:

- I. Preparation of new systems
    - 1. Cleaning
    - 2. Tightness
    - 3. Removal of moisture and air
    - 4. Charging.
- II. Operation of systems
    - 1. General operation
    - 2. Instrumentation
    - 3. Moisture
    - 4. Oil
    - 5. Purging.

I. Preparation of New Systems

1. Small or complete refrigerating systems of the package type are **cleaned** by the manufacturer, by the circulation through coils, shells and tubing of an agent for dirt elimination. Compressor parts are thoroughly cleaned during assembly.

In large systems, shells, coils and compressors come from the manufacturer free from dirt, and piping that is erected during installation should be thoroughly cleaned as erected. After the test for tightness is made, the entire system should be blown down to remove dirt and foreign matter.

2. Package equipment and factory-assembled systems are usually tested for **tightness** in conjunction with air and moisture removal (see Par. 3).

Large systems erected on the job should have the entire piping work tested with compressed air for leaks under pressures up to those specified for equipment using various refrigerants. These air pressures are obtained generally by use of the refrigerant compressor. When pumping a pressure on the system it is well to do so in stages rather than all at one time, in order to avoid excessively high air temperatures such as may cause explosion. If the pressure is raised to 50 psi and tested for major leaks, then brought up to the final test pressure in two or three more pumping operations, no trouble will be experienced. When the machine is used as an air compressor the usual amount of cooling water must be circulated.

After the system has been made tight under air pressure, all coils, coolers and pipe work generally should be **blown out** to remove dirt. The system should be tested under vacuum as a final check for leaks; when subjected to external atmospheric pressure, small pin holes closed by dirt while under internal pressure will often show up.

Before starting the compressor to pump up an air pressure, it is important that the cylinders be well **lubricated.** If the machine is fitted with a mechanical lubricator for supplying oil directly to the cylinders, oil may be pumped to the cylinders by the hand attachment. If the machine is not provided with such a lubricator, the cylinder heads should be removed and the cylinder walls well covered with oil.

Before starting the air test, open the stop valve on the compressor discharge pipe, and all other valves between the compressor and inlet valves to the various cooling coils or evaporators, and close the suction stop valve at the compressor inlet. Then open the valve on the suction inlet to the compressor between the suction stop valve and compressor to permit air to be drawn into the cylinders when the compressor is started. If no valve has been provided at this point it will be necessary to open a flange to provide the necessary air intake.

With the compressor in operation, pressure will be built up on the entire high side of the system, and finally brought up to 300 psi in the case of ammonia. After the high-side test is finished, the low side should be tested and made tight at about 150 psi. All joints, valve stems and the like are then tested again for leaks.

If the compressor is driven by either a steam engine or an internal combustion engine, it may be operated at slow speed during the air test operations. If driven by a constant speed motor, it will operate at full speed, building up the air pressure more rapidly; for this reason the pressure gage should be closely observed by a man stationed at the starting switch to stop the machine promptly when the pressure has been raised to the predetermined point.

After the air test is completed, the cylinders are opened for examination. The scale trap or strainer in the suction line at the compressor is also cleaned out.

3. Package equipment and factory-assembled systems are **evacuated** and **dehydrated** by the use of high vacuum pumping and heating by oven or other means so that the systems can be brought down in some cases as low as 50 microns (.001 millimeter absolute pressure). This procedure also serves as a tightness test.

Large plants erected on the job should have had most of any moisture removed during the blow-down following the test for leaks. Air removal is done by the use of a small portable pump-down compressor which usually evacuates to 25 in. of mercury and leaves the system commercially free of air.

4. Small factory-built systems and package equipment are **charged** with definite quantities of refrigerant, predetermined during the process of development for maximum efficient operation.

Large systems are charged with refrigerant usually introduced into the system through a charging valve and connections located on the outlet of the liquid receiver between the receiver outlet valve and low-pressure side of the system. A drum of refrigerant is connected to this charging valve and the compressor started. The drum-charging valve is then opened partly, as are also one or more expansion valves.

Liquid refrigerant will then flow from the drum through the valve, which is opened wide, charging connection and liquid line to the cooling coils, where it will be evaporated by the absorption of heat; then in the form of a gas it will pass through the suction line to the compressor from which it will be discharged to the condenser. As condensation occurs, the liquid will collect in the receiver, where its presence will be indicated by the receiver gage glass.

Since the outlet of the receiver is closed, none of this ammonia can pass out to the coils; therefore as the charging operations proceed it will continue to collect in the receiver until the system is fully charged. It is, however, a good plan to put only one or two drums in the system at first, then stop the compressor and make a careful check for possible leaks.

A good method of detecting ammonia leaks is to go over the entire system with lighted sulfur sticks, as any mixture of sulfur and ammonia gas will produce a white fog. In the case of brine tanks, the coils should be similarly tested before making brine. Leaks in ammonia condensers may be detected by the use of litmus paper held in the stream of the outlet water. The presence of ammonia in the water will be manifested by a discoloration of the litmus paper, white litmus paper turning red in water containing ammonia.

Testing for Freon leaks is done by means of the Halide torch.

During charging operations it is of course important that water be circulated over the condensers.

II. Operation of Systems

1. General operation.

a. Starting and stopping. Modern compressors are automatically started and stopped with predetermined temperature changes in the refrigerated rooms, usually of the vertical enclosed type, driven from an electric motor.

Where automatic control is employed, no opening or closing of the stop valves or bypass valve is necessary, since the motor, which is usually of the double squirrel-cage type having a high starting torque, will start the compressor without difficulty and without manual attention.

In the case of the synchronous motor-driven compressor, of either the belted or direct-connected type, special equipment provided for starting and stopping operations includes a push-button start and stop arrangement which makes it necessary only to push the starting button to start the compressor after the operator has closed the suction and discharge valves on the machines and opened the full size bypass. After the motor has been brought up to full speed, the discharge valve is opened while the bypass is closed, after which the suction stop valve is opened slowly and the load gradually put on the machine.

The same procedure is followed when starting other non-automatic compressors of either the vertical or horizontal type.

After the compressor is started and the suction stop valve is fully opened, the liquid outlet at the receiver is opened in order that the evaporator may be supplied with liquid.

In the case of manually controlled machines, the first step in shutting down is to close the main liquid outlet valve at the receiver; after the evaporating pressure has been pumped down to a few pounds above zero the compressor is stopped and the suction stop valve closed. This simple operation is all that is necessary for a small plant.

In larger plants, where two or more machines may be in use, one or more of them may be taken off the line by simply stopping the compressor and closing the suction stop valve.

b. General operating pressures. The operation of any given plant will depend of course upon the conditions to be met, but the fundamental principles of operation are the same for all types of installations.

The evaporating pressure, commonly called the **suction pressure,** is determined by the room and brine temperatures required. For some types of coils it is necessary to maintain a temperature difference of from 10 to 12 F between the room or brine temperature and the evaporating ammonia. Other types of evaporators, such, for example, as those used in the more modern freezing tank design, require a temperature difference of only from 2 to 3 F.

Similar variation is found also in rooms using evaporators over which the air is moved at high velocity by means of a fan. The reason for this difference in performance is that air, brine, or other media give up heat more rapidly when moved over the cooling surfaces rapidly. As previously stated, the operator must determine from experience what evaporating pressure is best suited to the particular conditions, but in any event this pressure should be carried as high as possible in the interest of power costs.

The amount of surface in any evaporator will have a direct bearing on capacity; in cases where the surfaces are insufficient, a much lower evaporating pressure will be needed.

The **condensing pressure** should be held as low as possible. It depends upon the amount and temperature of the condensing water available, and the total amount of cooling surface, as well as, to some extent, on the type of condensers used. Where liberal condensing surfaces have been provided, it should be possible to hold the condensing pressure to that corresponding to a temperature of about 8 to 10 F above the temperature of the water leaving the condenser.

2. For maintaining high operating performance, **pressure gages** and **thermometers** should be installed at several points throughout the plant and periodic checks of certain relations made. It is well to have pressure gages for both suction and discharge of the compressor and also thermometers at both places. Refrigerant gages can be obtained which have a temperature scale corresponding to the pressure scale. By comparing the temperature of the suction gas with the saturation temperature on the suction gage the superheat in the suction gas is obtained. When such superheat is from 5 to 15 F, a good condition prevails.

Where operation is at normal conditions and proper suction gas superheats prevail, an average normal discharge temperature results. Should the discharge thermometer show considerably higher temperatures than normal, there is trouble in the compressor due to leaky or broken valves or piston rings and an examination of the compressor is necessary.

With compressors equipped with forced lubrication, there should be a gage on the oil pump which when compared with the suction gage will show oil pressure. This pressure should be that specified by the manufacturer for satisfactory lubrication.

A pressure gage at the condenser gas

inlet and a thermometer in the condenser liquid outlet are used to check condenser performance by comparing the liquid temperature with the saturation temperature for the gage pressure at condenser inlet.

If the liquid temperature shows more than 20 F below the saturation temperature when the condenser is a free draining one, there are sufficient non-condensible gases present so that purging of the system is indicated.

Thermometers on the water inlet and outlet of the condensers will show any change in both initial and outlet water temperatures, which in turn will serve to indicate when the condensers are dirty and need cleaning. For example, suppose the inlet water temperature is 75 F and the outlet temperature 85. If, after a day's operation it is found that while the inlet temperature remains constant the outlet temperature continues to decrease and approach that of the inlet temperature, this is conclusive proof that the water surfaces of the condensers are becoming dirty. As a consequence, heat transmission has been so reduced that the ammonia is prevented from giving up its heat to the condensing water.

3. **Moisture** in a system may cause troubles such as copper plating or freezing in an expansion valve, particularly where the water is not soluble in quantity in the refrigerant. In the case of ammonia such moisture will form aqua ammonia in the low side and affect the boiling point.

Moisture may be admitted to a system when opening the low side for repairs, thus admitting humid air. The moisture content of the system may also be increased by an air leak on the low side or a very slight water leak in commercial equipment using a water-cooled condenser. In case of failure through ice formation in the expansion valve, a drier may be installed in the liquid line for a sufficient period of time to remove the moisture, if it has occurred from air being admitted to the system. However, an air leak or water leak will require the necessary repairs, dehydration of the system, and recharging with dry refrigerant and dry oil.

4. **Oil contamination** due to oil pumping of the compressors may cause inefficient performance of parts of the system.

A great portion of the oil leaving a compressor discharge may be returned to the crankcase from suitable oil separators installed in the discharge lines. However, oil separators are not 100% efficient in performance, and hence some oil passes on to the condenser and is carried with the condenser liquid into the low side or evaporator. When the refrigerant is such that oil is not miscible to any great extent, as is the case with ammonia, oil will collect in the low side and will have to be removed by draining.

If oil is miscible with the refrigerant it may return with the suction gas to the compressor, which is usually provided with means of returning such oil to the crankcase. Or if the evaporator will not return oil with suction gas, either draining or distilling may be used for its elimination.

In all cases oil accumulations in evaporators will affect their performance. Such performance in the case of refrigerants in which oil is not miscible will have effective surface reduced; and in the case where oil is miscible in the refrigerant, will raise the boiling temperature of the mixture.

5. When condenser performance indicates the presence of non-condensible gases, the system should be subjected to **purging**. In general, such non-condensible gases are air and can be removed by properly designed purging devices with which many plants are equipped.

In general, purging should be done at the proper point for economical and rapid elimination of the non-condensible gases; this point is not the same for all refrigerants.

When the air has a greater density than the refrigerant gas in the high side of a system, as is the case with ammonia, there is a definite tendency for the heavier air to be forced to the lowest point above the liquid level in the receiver when the plant is in operation; hence purging should be done just above the liquid level.

When the air has a lesser density than the refrigerant gas in the high side, as is the case with Freons, such air tends to concentrate in the top of condenser and purging should be done at this point.

Purging devices, generally designed to bleed small quantities of gas from the proper place in a system, are equipped with cooling means to condense out as much refrigerant as the suction pressure of the system will accomplish; the remaining gas, which is a high percentage non-condensible, is discharged to the atmosphere.

If no devices are installed, hand purging can be accomplished by attaching an empty refrigerant drum to the purge point with a quarter-inch line, bleeding into this drum until flow ceases, and then condensing as much refrigerant as condensate can be charged back into the system through a sight glass and then discharging the gases to the atmosphere. This process should be repeated until the liquid temperature of the system indicates the removal of non-condensible gases.

PART V

DOMESTIC AND COMMERCIAL SYSTEMS

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29. HOUSEHOLD REFRIGERATORS AND FREEZERS

THIS chapter deals with household refrigerators and freezers of the electrically driven mechanical type and the heat operated systems.

REFRIGERATORS

Yearly Sales

1. Refrigerator sales are highly seasonal with as little as 3% of the annual sales falling in October or November and as much as 16% in April.[1] Thus, a factory must have large storage facilities or operate under widely varying load conditions. The storage problem is further increased by the number of sizes and quality variations to appeal to various consumer tastes. **Table 1** shows the estimated **annual sales** and the average **retail price** of all refrigerators produced in the United States. **Table 2** breaks down the 1949 sales of the mechanical type for eleven National Electrical Manufacturers Association companies. **Table 3** shows the yearly percentage sales of refrigerators in the 6 cu ft and under group, and in the 7 cu ft and over group. It is interesting to note the very marked shift from the smaller to the larger sizes.

Design Trends

2. In general, the refrigerator user of today assumes satisfactory mechanical operation and places greater emphasis on improved **appearance, utility,** and **convenience.** As a result, the most significant design trends over the last few years have been in these categories.

Table 1. World Household Refrigerator Sales of U. S. Manufacturers

Year	Unit sales	Avg retail price	Retail value
Up to 1920	10,000	$600	$ 6,000,000
1921	5,000	550	2,750,000
1922	12,000	525	6,300,000
1923	18,000	475	8,550,000
1924	30,000	450	13,500,000
1925	75,000	425	31,875,000
1926	210,000	390	81,900,000
1927	390,000	350	136,500,000
1928	560,000	334	187,040,000
1929	840,000	292	245,280,000
1930	850,000	275	233,750,000
1931	965,000	258	248,970,000
1932	840,000	195	163,800,000
1933	1,080,000	170	183,600,000
1934	1,390,000	172	239,080,000
1935	1,688,000	166	280,208,000
1936	2,180,000	164	357,520,000
1937	2,500,000	171	427,500,000
1938	1,410,000	172	242,520,000
1939	2,085,000	169	352,365,000
1940	2,720,000	154	418,880,000
1941	3,700,000	160	592,000,000
1942	520,000	—	—
1945	263,860	—	—
1946	2,100,000	207	434,700,000
1947	3,400,000	255	867,000,000
1948	4,766,000	260	1,239,160,000
1949	4,450,000	255	1,134,750,000

Table 2. Household Refrigerator Sales Statistics for National Electrical Manufacturers Association Companies, for Year 1949

Sizes	Domestic (48 states and D.C.)	Canadian	Other Foreign	Total
1. Less than 4 cu ft	2,208	—	54	2,262
2. 4 cu ft	25,130	—	2,824	27,954
3. 5 cu ft	—	—	—	—
4. 6 cu ft	504,886	2	39,796	544,684
5. 7 cu ft	879,198	29	33,809	913,036
6. 8 cu ft	1,157,030	5	44,314	1,201,349
7. 9 and 10 cu ft	978,485	2	23,224	1,001,711
8. 11 and 12 cu ft	157,297	—	4,091	161,388
9. 13 cu ft and up	35	—	1	36
10. TOTAL	3,704,269	38	148,113	3,852,420

3. While the demand for more **refrigerated food storage** space has increased, the kitchen size in homes has decreased. As a consequence, there has been a marked increase in the ratio of the food storage volume of the refrigerator to the floor area occupied by it. This has been accomplished by reduction in the physical size of the **motor compressor** unit, and relocating it, so that all or a portion of the bottom of the food compartment can be lowered (Fig. 1).

4. The greatly increased use of frozen foods has resulted in the need for providing refrigerators with much larger and colder **frozen food storage compartments.** In a large number of refrigerators this increased frozen food storage volume is provided by a freezing compartment which extends substantially the full width and across the top of the food compartment (Fig. 1). So that the **evaporator** can be operated at the low temperatures required for proper frozen food storage, some form of insulated baffle is placed beneath it to separate it from the food compartment proper. This baffle is so designed as to permit sufficient air movement around it

Table 3. Refrigerator Sales, by Size of Box (1934–1949)

Year	6 cu ft & under	7 cu ft & over
	%	%
1934	78.1	21.9
1935	83.5	16.5
1936	85.2	14.8
1937	83.9	16.1
1938	84.4	15.6
1939	90.3	9.7
1940	90.2	9.8
1941	80.1	19.9
1946	9.3	90.7
1947 (9 mos.)	11.4	88.6
1948	12.9	87.1
1949	15.0	85.0

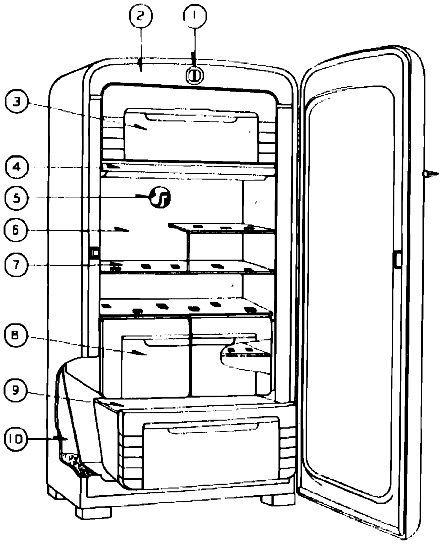

Fig. 1. Refrigerator Cabinet with Across-the-top Freezing Compartment

1. Temperature control knob
2. Breaker strip
3. Frozen-food compartment
4. Insulated baffle and meat tray
5. Food compartment light
6. Bottle storage space
7. Glass shelf
8. High-humidity compartment
9. Vegetable storage drawer
10. Unit compartment

to provide the necessary cooling for the main food storage compartment.

Another approach to the problem of large and adequately cooled frozen food storage compartments in refrigerators has been the introduction of a two-door design **(Fig. 2)**. In this modification there are two compartments completely separated from each other and each has its own exterior door. Both compartments are cooled by a single refrigerating unit. The generally accepted method is to cool the frozen food compartment directly with the unit and refrigerate the main food storage compartment by means of a secondary system attached to the primary one (Fig. 5). In some of the two-door refrigerators, independent temperature control of each compartment is provided.

5. The trend toward specialized interior appointments has been marked. **Meat storage compartments** maintained at temperatures slightly above freezing with relative humidities of 90% are common. Special **high humidity** compartments for storage of leafy vegetables are provided in practically all refrigerators. **Special compartments** with various heating means for maintaining butter at a temperature convenient for spreading are appearing on more and more of the refrigerators offered to the public (Fig. 2).

6. Recent years have seen the introduction and increased use of refrigerators in which all or a substantial volume of the main food storage compartment is maintained at **high relative humidities.** The purpose is to reduce the rate of evaporation from the surface of the food stuff to such an extent that they may be stored in uncovered vessels for considerable periods of time without deleterious effects (Figs. 1 & 2).

7. Bacteria and mold increase as food is maintained in a more moist condition, thus, while high humidity retards drying, it tends to hasten mold and bacterial deterioration. To retard this bacteria and mold growth, many domestic refrigerators are being equipped with **ultra-violet lamps.** The major portion of the radiation from these lamps is in the bactericidal region of the ultra-violet spectrum. These lamps also produce small quantities of ozone. In addition to their bactericidal action, they are also effective in reducing cabinet odor level and odor transmission from strong to sensitive foods. To reduce the odor level, activated charcoal **odor adsorbers** are also used. These adsorbers can be reactivated at a temperature of from 375 to 475 F every few months and replaced once a year.

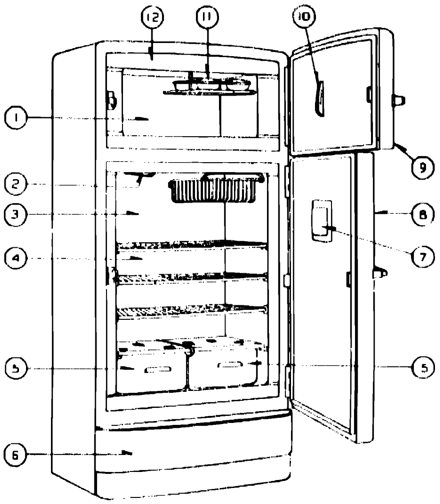

Fig. 2. Two-Door Refrigerator Cabinet

1. Frozen-food compartment
2. Food compartment light
3. Bottle storage space
4. Food compartment
5. Vegetable storage drawers
6. Unit compartment
7. Butter storage compartment (heated)
8. Door—food compartment
9. Door—frozen-food compartment
10. Thermometer
11. Ice-trays
12. Breaker strip

8. A very important feature that has been recently introduced in domestic refrigerators is **automatic defrosting** of the evaporator. Early approaches to the problem of removing the frost from the evaporator were confined to providing a manual means for modifying the operation of the standard temperature control. This modification permitted the evaporator to operate above 32 F for a period of time. More recent practice has been the introduction of a clock mechanism which daily interrupts the power supply to the refrigerator for a sufficient length of time to permit the day's accumulation of frost to melt from the evaporator during the early morning hours. However, these schemes present the problem of maintaining temperatures adequate for the proper storage of frozen food, ice cream, etc.

Another approach has been to separate the sub-freezing compartment from the main food storage compartment and operate the evaporator for the latter above freezing (Fig. 2). The sub-freezing compartment has to be defrosted by the usual methods, although less frequently than the conventional type.

More recently, there have been introduced rapid automatic defrosting systems in which heat is applied to the refrigerating systems to increase the rate at which the frost can be melted so that the defrosting time is materially shortened **(Fig. 3)**. The defrosting period is of from 5 to 30 minutes' duration. This short defrosting cycle should permit no substantial rise in the frozen food temperature or in the food compartment temperature. These types of units permit the safe storage of frozen foods over considerable periods of time, at the same time keeping the evaporator relatively free from frost. These refrigerators also provide means for evaporating the defrost water into the ambient air so that a completely automatic defrosting refrigerator is provided.

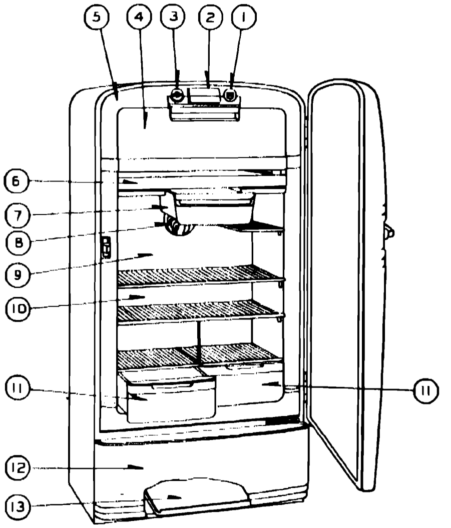

Fig. 3. Refrigerator with Automatic Rapid Defrost

1. Automatic defrost control
2. Butter storage compartment
3. Temperature control knob
4. Frozen-food compartment
5. Breaker strip
6. Drain pan and baffle
7. Meat storage drawer
8. Sterilamp
9. Bottle storage space
10. Food compartment
11. Vegetable storage drawers
12. Unit compartment
13. Water evaporating pan

Cabinet Design

9. The fundamental factors in the design of a cabinet are its usable food storage **capacity** in cubic feet and its external **dimensions,** both of which are influenced by competitive consideration. A method of computing useful cubical contents and useful shelf area is contained in the American Standards Association publication B38.1-1944.

10. The **external shell** of the cabinet is generally a single fabricated steel structure which supports the inner **food compartment, the door,** and the **refrigerating mechanism.** In general, the door and **breaker strip** construction is similar to that shown in **(Fig. 4)**. The door covers the whole front of the cabinet and the plane of the breaker strips is parallel to the back wall. A single plastic sheet serves as the inner surface for the door and the door breaker strips. Mechanically, the overall **structure of the cabinet** should be strong enough to stand shipping. A cabinet that will ship is almost certain to be strong enough to stand daily usage. The inner food liner should be so designed that it does not chip or craze at the points of support. Shelves must be designed with

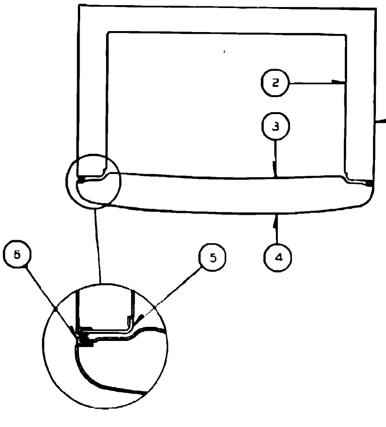

Fig. 4. Door and Breaker Strip Construction

1. Outer shell
2. Inner liner
3. Plastic inner door pan
4. Door
5. Breaker strip
6. Door gasket

adequate strength so that they will not sag under the heaviest load and the cabinet hardware should be satisfactory for at least 50 door openings per day for a period of perhaps 15 years.

11. **Insulation** in a domestic refrigerator varies from 2 to 3½ inches in thickness and the greater thickness ordinarily is used with the large size cabinets. Insulating materials in common use are rock wool, glass wool, balsam wool, corrugated paper, etc. Although it is desirable to obtain as nearly a vapor-tight exterior cabinet shell as possible, consideration should be given the fact that when water vapor is migrating from the ambient air into the insulation, it is also passing from the insulation into the area around the cold evaporator. Consequently, the design of door gaskets, breaker strips, and other parts should provide maximum barriers for vapor flow from the room air to the insulation, but as free a passage as is practicable between the insulation and the inside of the cabinet.

The effect of insulation thickness on heat leakage is not the only important consideration. Another very important one is maintaining the exterior of the surface of the cabinet at a sufficiently high temperature to prevent moisture condensation. Areas where condensation usually occurs will be around the hardware or along the edge of the door opening. Good practice suggests that with a 40 F refrigerator interior temperature in an 80 F room, no external surface temperature on the cabinet should be more than 7 F below the room temperature.

12. In the early days, **cabinet finishes** presented one of the most serious problems because they deteriorated rapidly. However, modern synthetic baked enamel finishes give an excellent account of themselves. Customarily, they are applied over a chemically treated surface which gives better adhesion to the paint and prevents spreading of the rust if the finish is scratched through. In addition to the synthetic enamel finishes there are improved vitreous enamel finishes that usually consist of one ground coat and two or three finish coats, each of which is fused on separately. The finish used almost universally on the food compartment liner is a white vitreous enamel. However, suitable synthetic organic finishes are now available for this application and some refrigerators on the market are using this type of finish on the food compartment liners.

13. **High humidity compartments** are used to retard evaporation of moisture from uncovered foods. In conventional refrigerators the humidity in the food storage compartment is low, due to the presence in the cabinet of the cold evaporator. Rapid loss of moisture from the uncovered food results from this low humidity condition. The drying effect can be greatly reduced by cooling the food storage compartment by means of a large cooling surface operated at a few degrees below the food compartment temperature.

These high humidity compartments may be as small as a single drawer for the storage of meats or vegetables, or they may include the whole above freezing storage space of the refrigerator. The large compartment may be designed for only moderately high humidities for the storage of leftovers, and drawers provided for the very high humidities needed for the storage of leafy vegetables. If the small-sized drawer compartment is intended for the storage of meat it should be placed in a location where the temperature obtained in it can be at or near 32 F (Fig. 3). If the drawer is intended for vegetable storage it may be placed anywhere in the cabinet, but it is usually located at the bottom.

The humidity in a closed container will be high because the cooling surface is the whole surface of the drawer, operating at practically the same temperature as the air within the drawer. This same principle can be extended to larger sized compartments by providing an enclosed compartment two or three shelves high within the cabinet, with sufficient clearance between the walls of the two compartments to permit adequate circulation of cold air around the enclosed one (Fig. 1).

14. When the main food storage compartment is operated at high humidity, it is generally accomplished by separating the low temperature evaporator from the main storage compartment by suitably insulated baffles, and cooling the food compartment by coils attached to its sides

adjacent to the insulation. These coils are usually the evaporating portion of a closed **secondary refrigerant circuit,** the condensing portion of which may be tubes in contact with the primary evaporator, or some tubes or a tank cooled by a separate primary loop buried in the insulation. A secondary system is shown in **Fig. 5.** The secondary circuit is charged with a volatile fluid, usually the same as is used in the primary circuit. As the liquid refrigerant in the tubes on the food compartment walls evaporates, it passes upward to the secondary condenser, where it condenses, and then returns to the lower tubes to repeat the cycle. The secondary condenser section is removably coupled to the main cooling surface, so that when the primary refrigerating unit is removed, the secondary circuit can remain in the cabinet.

This secondary cooling system has two main advantages over the one in which part of the primary circuit is used for cooling the high humidity compartment. In the case of unit failure, there is no need to remove the high humidity cooling circuit and the same unit can be used in refrigerators that do not employ high humidity sections. High humidity compartments have also been cooled by large areas of finned coils located within the food compartment, but this design is not commonly used.

High humidity compartments may have their own inner doors (Fig. 1) or they may be sealed by a single main cabinet door. In the latter case, the partition separating the low from the high humidity compartments usually seals against the door with a rubber gasket. A third arrangement employs two outside doors, one closing the low temperature compartment and a large one covering the opening of the high humidity main food storage compartment (Fig. 2).

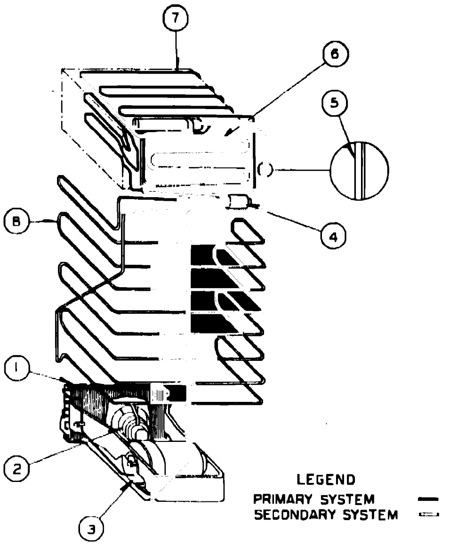

Fig. 5. Refrigerator with Secondary System for Food Compartment Cooling

1. Tube and fin condenser
2. Fan
3. Compressor
4. Secondary system control valve
5. Suction line and capillary tube in heat exchange relation
6. Secondary condensing tubes clamped to back of frozen food compartment
7. Frozen food compartment cooling tubes
8. Secondary tubes cooling food compartment

In the wintertime, the low ambient dewpoint makes it difficult to maintain a high relative humidity in the refrigerator. This condition can be helped by a design in which a large cooling area serves to maintain a very small temperature difference between the inside surfaces and the air in the food compartment. However, in the summer when a high ambient dewpoint is encountered, excessive wetness within the food compartment may result. Each time the cabinet door is opened, moisture is deposited on all of the walls, shelves, and dishes in the refrigerator. As the design of this type of food storage compartment is such as to reduce evaporation from food, evaporation from all other surfaces is also very low and the moisture collected, due to repeated door openings, may result in drippage.

15. While the bactericidal effects of **ultra-violet** and **ozone** have been known for a long time it is only recently that suitable lamps for use in domestic refrigerators have been available. A lamp must have sufficient ultra-violet output, deliver a controlled amount of ozone, and yet, give off very little heat. Lamps are available that radiate the major portion of their energy at the most effective bactericidal wave length, 2537 Ångstrom units. These lamps are most needed in areas operating at relative humidities above 60%, with food in uncovered dishes, since at lower relative humidities, food will dry out so rapidly that no mold or slime will occur on the surface. As the germicidal effect which occurs in the areas shaded from the radiation from the lamp is produced partly by the ozone generated by it, the lamp should be placed so that any circulation will tend to carry the ozone throughout the food storage compartment (Fig. 3).

16. If the cabinet is to be a component of an automatic defrost refrigerator, adequate provision must be made to take care of the defrost water. Suitable means must be provided to catch the defrost water from the evaporator. A container may be provided within the refrigerator to collect the defrost water, and must be emptied at regular intervals. If the defrost water is to be automatically disposed of, some sort of a drain pipe must be provided to convey the water from the tray beneath the evaporator to some external container from which it can be evaporated (Fig. 3). The container should be of sufficient size to take care of the maximum amount of water that may be discharged at any one defrosting period and if natural evaporation is to be the means for disposing of the water, the pan should be of ample capacity to take care of periods when the rate of frost collection may exceed the rate of evaporation from the pan. The container should be made of suitable material to resist corrosion as it will be subject to deterioration not only from the water, but also, from acid-containing foods which may inadvertently find their way into the pan.

Design of Mechanical Unit

17. Practically all household refrigerators employing a mechanical compressor utilize **hermetic units.** In this design, the motor and compressor are sealed into a gas-tight housing, eliminating the shaft seal, belt, pulley, and open motor bearings. The motor compressor unit is built as a single integral assembly and enclosed in a steel shell (Fig. 5). Two **refrigerants** are used in present production of hermetic units, **Freon-12** (dichlorodifluoromethane) and **Freon-114** (dichlorotetrafluoromethane). Freon-12 is used in both **rotary** and **reciprocating** compressors and Freon-114 in rotary compressors only.

18. Refrigerant **pumps** are either **reciprocating** or of positive displacement **rotary** design. If reciprocating, the compressors are either of the automotive type with the wrist pin in the piston or a scotch yoke type **(Fig. 6** and **Fig. 7).** The advantage of the latter is the flexibility between the axis of the piston and the motor axis. With the automotive type, it is necessary that the wrist pin be exactly at right angles to the piston axis, otherwise noise will be introduced due to the misalignment of parts. When parts are not built with sufficient rigidity, misalignment under load may occur even though no-load accuracy exists. This condition does not exist in the scotch yoke design, because it has flexibility in two directions.

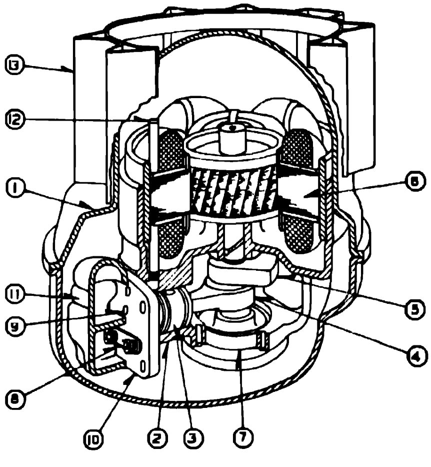

Fig. 6. Hermetic Motor Condenser Unit with Reciprocating Pump

1. Hermetic housing
2. Cylinder
3. Piston
4. Connecting rod
5. Counter balance
6. Motor
7. Outer bearing
8. Discharge valve assembly
9. Suction valve port
10. Valve plate
11. Cylinder head
12. Suction gas intake pipe
13. Cooling fins

19. Rotary pumps (Fig. 8) are generally either of two types. In one type the integral rotor and shaft assembly is concentric with the circumference of the rotor and eccentric to the circumference of the cylinder. Rectangular blades are set into slots in the rotor which bear out against the circumference of the cylinder by

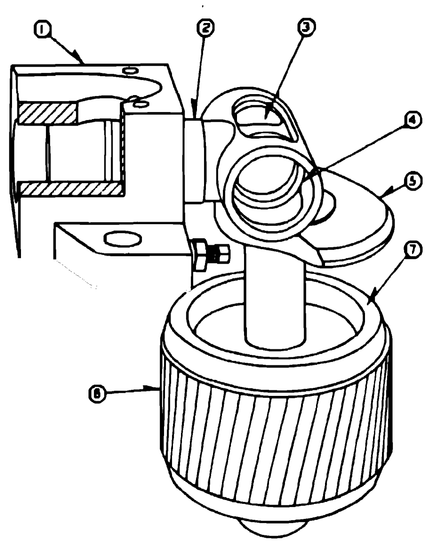

Fig. 7. Scotch Yoke Compressor

1. Cylinder
2. Piston
3. Crank pin
4. Cross slide
5. Counter balance
6. Rotor punchings
7. Die cast conductors

means of gas pressure, springs, or centrifugal force; and as they move around the circumference, the gas trapped between them is compressed. The other type employs a shaft and an enlarged eccentric assembly that is concentric with the cylinder wall. On the eccentric is a cylindrical rotor of such size that it operates at one point close to the cylinder wall. A blade operates in a slot in the cylinder wall and is held in contact with the rotor by means of springs or gas pressure to provide a seal between the high and the low pressure sides. As the eccentric rotor revolves, gas is drawn in just after passing the blade and is then compressed and discharged at the other side of the blade. A modification of this latter design employs a plate fastened to a cylindrical piston concentric with the rotor. This plate oscillates in a slot in the cylinder wall between two rockers.

20. The **valves** for refrigerant pumps are ordinarily metal discs or reeds bearing on a valve seat. As valves are a prolific source of noise and inefficiency, much ingenuity has gone into their design.

21. The compressor **motors** used are of the squirrel-cage induction type. They may be either a straight split phase design **(Fig. 9)** or they may employ a capacitor in the auxiliary winding for starting, running, or both. For connecting the auxiliary winding into the starting position an external magnetic or thermal switch is used **(Fig. 10)**. For 50 and 60 cycle operation, 4 pole motors are used and on 25 cycle 2 pole motors are employed. The rotors employed have built-up copper or die cast aluminum conductors. The starting windings are insulated with two or three layers of impregnated cotton or enamel. If enamel is used, it is imperative that it be not soluble in the lubricating oil or refrigerant that is used. All of the other motor insulating materials, such as cell slot insulation, must be of such a nature that they are unharmed by the lubricating oil or refrigerant. It is also important that these fluids do not dissolve any substance out of the insulation that will cause trouble in the system.

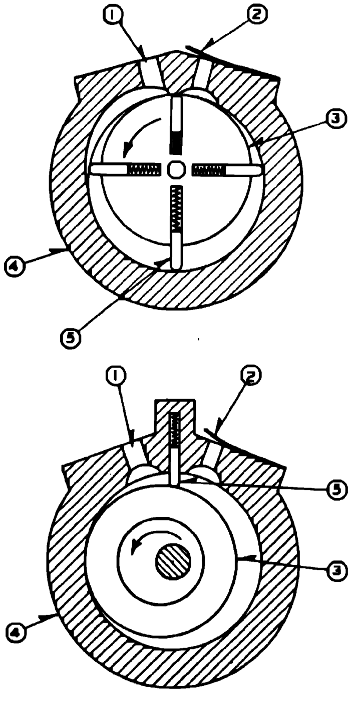

Fig. 8. Rotary Pumps

1. Intake port
2. Discharge port and valve
3. Rotor
4. Cylinder
5. Blade

22. When applying a motor to a domestic refrigerator compressor, design changes can be made in the associated equipment to give desirable motor loading. In this respect, the situation is different from the general motor applications. There are three points that must be checked in the application of a motor to a refrigerator compressor. First the motor should be capable of carrying, without **overheating,** the maximum possible load that may be imposed steadily on the machine. This load may be obtained by operating the refrigerator in an ambient of 110 F with the refrigerator door open.

Secondly, the motor should have sufficient **maximum torque** to carry the heaviest momentary load at the lowest possible voltage. Such a load might be achieved by placing the refrigerator in a 110 F room with the unit inoperative and the cabinet door open to allow all parts of the refrigerator to stabilize at 110 F and then with the cabinet door closed, operating the refrigerating unit to pull the cabinet temperature down. While the maximum load under this condition may be heavier than the one previously referred to, the temperature rise of the motor need not be considered because of the low thermal capacity of the modern evaporator permitting the machine to pull through this maximum load quickly. With the voltage throughout most of the United States ranging between 115 and 120, a machine which will pull through this maximum point at a voltage of between 100 and 105 volts is probably safe.

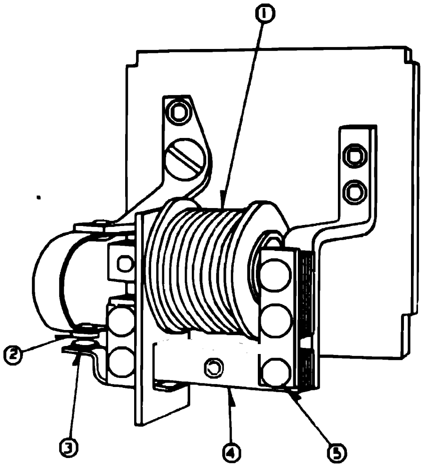

Fig. 10. Starting Relay

1. Coil
2. Stationary contact
3. Movable contact
4. Armature
5. Pivot

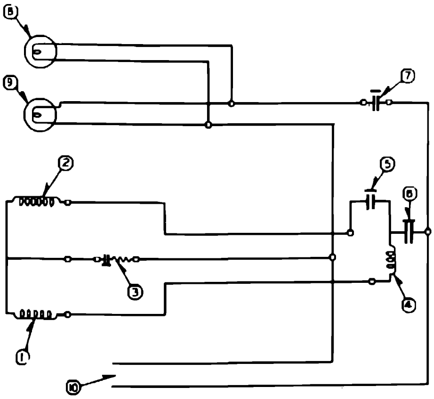

Fig. 9. Unit Wiring Diagram

1. Main winding of split-phase motor
2. Starting winding of split-phase motor
3. Motor protective device
4. Starting relay coil
5. Starting relay contacts
6. Thermostat contacts
7. Door-operated light switch
8. Cabinet light
9. Dial light
10. Power source

The third requirement for the motor is that it should have sufficient **starting torque** to start the compressor at 100 volts. In the unit designs using a capillary tube type of restrictor, and not employing an unloader, the starting torque is tied in with the degree to which the pressure between the high and the low side has an opportunity to equalize during the off period. Necessary starting torque is thus a function of control differential as well as capillary restrictor design. If the motor protection is an automatically reclosing device, it is not serious if the motor occasionally fails to start due to insufficient pressure equalization. If the protective device is not automatically resetting, it is imperative that the motor be able to start under all normal operating conditions. If the machine is not equipped with the type of restrictor that equalizes the pressure during a shutdown, then the motor must have enough torque to start the pump with maximum pressure differential across the piston. In this latter design, an unloader can be used to vent the pressure over the piston during the starting interval and reduce the starting torque required.

Condensers

23. The two classes of **condensers** are those of large mass and dimensions, cooled by radiation and thermal-convection air circulation, and the smaller type with forced-air circulation. Two common methods of building the large **plate type condenser** are employed. In the first, two sheets are roller welded together to provide refrigerant passages between them. The other employs a serpentine steel tube copper brazed or resistance-welded to a single steel sheet **(Fig. 11).** Two advantages of the plate condenser are that it is not necessary to employ a separate motor to drive the fan and it gives practically no trouble from the standpoint of dirt collection. The smaller type condenser (Fig. 5) employing a motor driven fan and finned tubing has a decided advantage of lightweight compact design and more satisfactory carrying of prolonged heavy loads. It has the disadvantage, however, of requiring an additional motor to operate the fan. The fan motor can be eliminated if the condenser size is increased considerably so as to obtain sufficient cooling by thermal-convection air circulation.

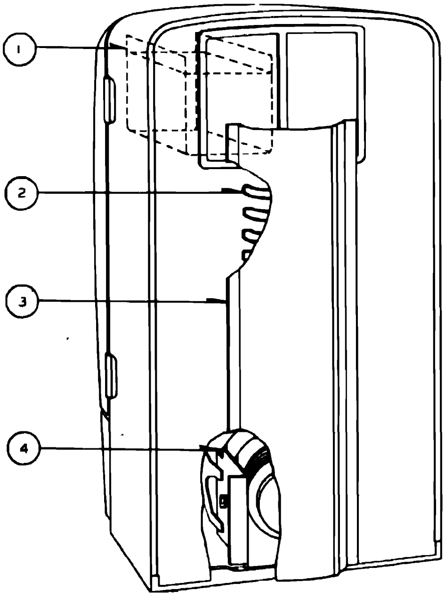

Fig. 11. Plate Type Condenser

1. Evaporator
2. Condenser tube
3. Plate condenser
4. Hermetic motor compressor unit

Expansion Devices

24. The type of **expansion device** now used almost entirely in domestic refrigerators is the **fixed orifice** type. It consists of a long **capillary tube** of internal diameter in the neighborhood of .040 inches, or an equivalent restriction. This fixed restrictor has the advantage of extreme simplicity and no moving parts, and lends itself exceedingly well to being placed in heat exchanger relation with the suction line, thereby effecting considerable gain in the unit capacity, particularly when using a Freon refrigerant (Fig. 5).

A disadvantage of this type of restrictor is the loss in capacity of the unit under some operating conditions. When the machine stops, some compressed gas passes into the evaporator, causing a loss in capacity. Even with the machine running, there is only a certain set of suction and discharge pressures under which any particular capillary tube will not pass gas or will not permit liquid to back up into the condenser. Extreme care must be exercised in the building of domestic refrigerating machines employing a capillary tube to be sure that the interior is kept scrupulously clean. If this is done and no water, wax, or other materials that harden at low temperatures are present, a fixed restrictor tube will not be troubled with plugging.

Evaporators

25. **Evaporators** are usually shaped in the form of a plate, U, or box (Fig. 11) and may be made by fastening tubing to sheets or by forming the refrigerant path with drawn convolutions in two sheets which are subsequently brazed or welded together. Evaporators now are so designed that the amount of refrigerant contained in them is approximately 1 lb. This has the advantage of reducing the severity of pulldown and also reducing cost. The refrigerant path is usually from the restrictor through the shelves, if any,

through the evaporator body and into a header of some type. If the evaporator is to be used with some type of secondary system for cooling the high humidity food compartment, consideration must be given to suitable surfaces for contacting the secondary condenser.

Ingenious ways of putting the evaporator into the cabinet without breaking the refrigerant lines have been developed. Evaporators are commonly placed through openings in the back or top, or the refrigerant lines may be run under the breaker strip and across the top. Whatever design is employed, care must be exercised to prevent cold spots which cause sweating and to seal the outside shell of the refrigerator cabinet to prevent air infiltration into the insulation.

26. If rapid automatic defrosting of the evaporator is to be employed, consideration must be given to some means for rapidly supplying heat to it. Heaters can be fastened to the cooling element or valves may be introduced into the refrigerant circuit that will bypass the restriction device and permit the hot gas from the compressor to discharge directly into the evaporator or a reversing valve may be employed which will reverse the normal functioning of the system and permit the evaporator to act as the condenser and vice versa. One system which has been recently introduced (shown in **Fig. 12**) eliminates the necessity for attaching heaters to the evaporator and the introduction of mechanical valves into the refrigerating system. In normal operation the container will operate with the refrigerant level up to the orifice. The refrigerant discharged from the capillary tube will pass through this orifice to the evaporator. During defrosting, the heater is energized, evaporating the liquid refrigerant discharged by the capillary tube. The orifice size is so designed that the pressure drop across it when passing this quantity of gaseous refrigerant is greater than the static head represented by the height of the inverted U tube. This results in the pressure building up in the tank sufficiently to empty the refrigerant from it via the inverted U tube. This extra charge circulating in the system floods the low side back to the second heat exchanger which loads the unit and raises the suction pressure. The refrigerant vaporized by the heater then enters the evaporator where it condenses, giving up its heat and melting the frost from the evaporator.

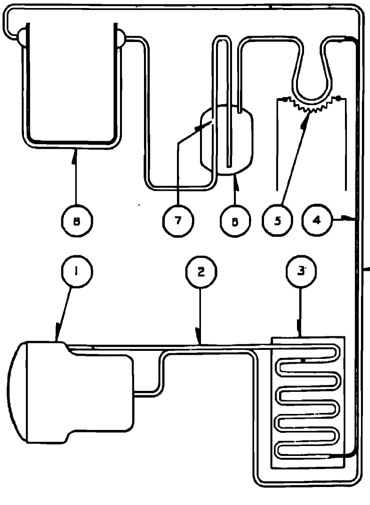

Fig. 12. Defrosting Method Using No Valves

1. Compressor
2. Heat exchanger
3. Condenser
4. Capillary tube
5. Heater
6. Storage tank
7. Orifice
8. Evaporator

Controls

27. Controls to start and stop the compressor are operated by a volatile liquid charged bellows system **(Fig. 13).** The temperature responsive element may be mounted in intimate contact with the evaporator and thus maintain a fixed average evaporator temperature; or it may be arranged in such a manner as to be responsive to both the evaporator and the cabinet temperatures. In the former method, the box temperature tends to fluctuate with changes in the ambient temperature. This fluctuation is reduced in the latter schemes. The latter scheme has the advantage from the standpoint of normal food preservation, while the former is better adapted to maintain constant frozen food temperatures under fluctuating ambient temperatures.

Systems employing auxiliary valves in the secondary system or automatically controlled dampers in the cold-air stream allow independent control of the frozen food and cabinet temperatures.

28. A **starting relay** to energize the auxiliary starting winding of the motor must be provided external to the hermetic compressor unit since it is not practical to use a centrifugal switch on the motor shaft. This is normally a magnetic or a thermal contact making device (Fig. 10). In the magnetic type the high starting current through the main motor winding moves the starting relay into the starting position. Then, as the motor speeds up, and the current drops off, the relay moves into the running position. While, at constant voltage a motor starting under no load has a current range of about 5 to 1 between the locked rotor current and the running current, it is necessary for these relays to operate between other more narrow current limits, in order to make contact under reduced voltage starting, and to break contact after starting under heavy loads and high voltage.

The thermally operated starting switch is essentially a time delay device. In the unenergized position, the starting winding contacts are closed. When the main contacts are closed, current to the running winding heats the thermal element at a rate which will cause it to open the starting winding contacts after the motor has attained proper speed. The normal running winding current is sufficient to keep the switch in the open position. Some of these switches are built to also serve as an overload protection device. Excessive running winding current through the element will open an additional set of contacts, interrupting the power supply to the motor.

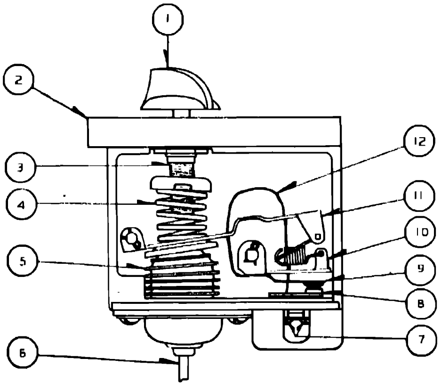

Fig. 13. Temperature Control Switch

1. Control knob
2. Case
3. Temperature control screw
4. Temperature control spring
5. Bellows
6. Bellows tube
7. Terminals
8. Stationary contact
9. Movable contact
10. Contact arm
11. Rocker arm
12. Flexible conductor

29. Some type of **motor protection** is ordinarily provided. The protective device usually possesses thermal lag that allows the motor to carry brief overloads without tripping out. There are two general types of overload devices, one is mounted on the compressor and is responsive to motor-compressor unit temperature and the motor current, and the other functions only in response to the motor current. The former arrangement has the advantage of being responsive to actual temperature of the machine. The protective device may be manual or automatic resetting.

30. For **defrosting** of the evaporator, the cabinet temperature control may be equipped with an additional position which will allow the evaporator to operate at a temperature above 32 F. Another method is to provide a manual means for adding additional force to the operating arm in such a manner that the turn-on temperature of the control is raised to a temperature above 32 F. After this operation of the control, the bias is automatically removed and it returns to its normal operating range.

An automatic **defrost control** consisting of a clock motor with suitable switching can be used. It can be set to initiate the defrosting period at some predetermined time and terminate it after some predetermined time interval. Another modification may be used in which the clock is the device used to initiate the defrosting cycle, and a temperature operated control to terminate it.

The defrost control can be complete within itself with the initiation of the defrost cycle determined by some operating function of the refrigerator, such as the frequency of operation, total time of operation of the refrigerating unit, or in response to predetermined number of door openings, etc. Termination of the defrost cycle may be actuated by a bellows assembly within the control. The temperature sensitive element can be fastened to the evaporator and it can be calibrated to terminate the cycle at some temperature above 32 F. Such a control is most practically used in a rapid automatic defrost system where the defrost time is of such short duration that it does not interfere with normal operation of the refrigerator. In considering the frequency of defrosting, attention must be given to the rate of frost collection on the evaporator. Recent reports[3] indicate that frequency and duration of cabinet door openings and ambient dew point are the major factors that influence the rate of frost collection on the evaporator.

Processing

31. **Processing** of the completed refrigeration unit after assembly includes extraction of the moisture from the motor winding and associated parts, removal of noncondensible gases, and charging the system with refrigerant and oil.

32. Thorough **drying** of the system is essential. An improperly dried system will be troubled with ice collection in the restrictors, and corrosion trouble. Drying is usually done by heating the unit to a temperature above 212 F for several hours and removing the water vapor by evacuating the unit or passing dry air through it. Care must be taken to remove all air before charging.

33. **Oil** and **refrigerant** are measured by either volume or weight methods. Modern machines with their extremely low refrigerant charge must be charged with considerable accuracy. Charging methods that involve measuring a certain volume by means of lowering the level in a sight glass or the use of a flow meter are accurate, but the charge cannot be checked in case of doubt. Oil and refrigerant of high purity in regard to moisture and noncondensible gases are essential.

Performance

34. The performance of domestic refrigerators refers to various aspects of the refrigerator behavior under different conditions of room temperature and humidity, internal temperature of the cabinet, and various conditions of loading. The data obtained are usually:

a. Rate of consumption of fuel or electrical energy
b. Per cent running time of the compressor unit
c. Temperatures obtained in the cabinet interior including the specialized compartments and freezing compartment
d. Characteristics of auxiliary functions such as production of ice cubes
e. Reaction of box to high relative humidity such as condensation on the surface of cabinet and adsorption of water into the insulation
f. Resistance of the finish to light, scratching, grease, etc.
g. Mechanical durability of the mechanism and hardware.

In a mechanical refrigerator, a change in the ambient temperature has a marked effect on **energy consumption,** assuming the box temperature remains constant Three factors are involved as the ambient temperature increases:

(1) The differential through the cabinet wall increases and thus the load on the cooling unit increases. (2) More energy is required because of the greater temperature difference between which heat is absorbed in the evaporator and rejected to the air. (3) The efficiency of the system decreases because of more leakage past the

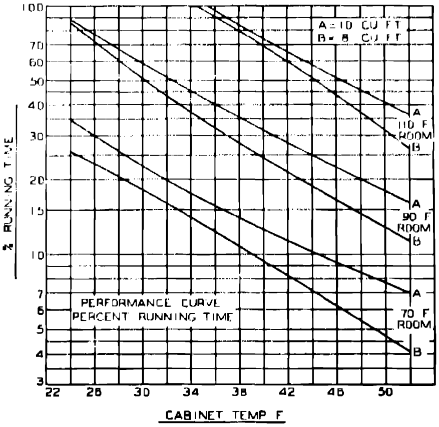

Fig. 14. Percentage Running Time vs Room Temperature and Box Temperature for Typical 8-cu ft and 10-cu ft Refrigerators in One Manufacturer's Line

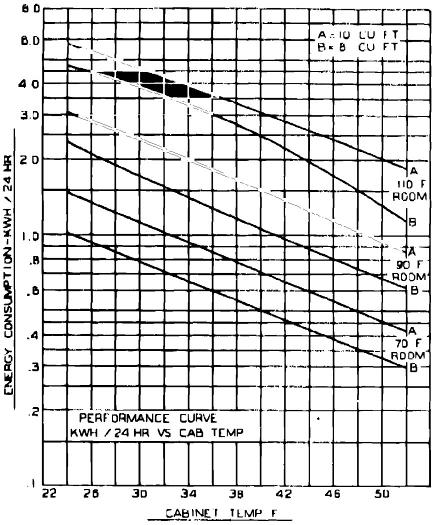

Fig. 15. Kilowatt Hours Consumption for 24 Hours for Typical 8-cu ft and 10-cu ft Refrigerators in One Manufacturer's Line

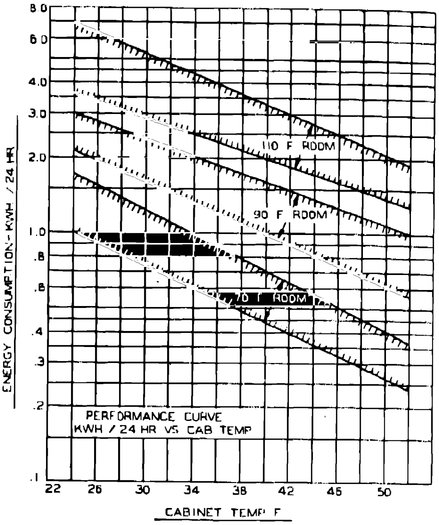

Fig. 17. Kilowatt Hours per 24 Hours for Refrigerators Produced by Nine Different Manufacturers during 1939, 1940 and 1941

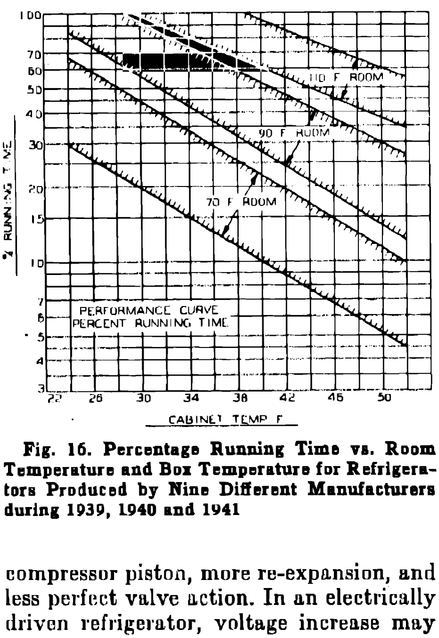

Fig. 16. Percentage Running Time vs. Room Temperature and Box Temperature for Refrigerators Produced by Nine Different Manufacturers during 1939, 1940 and 1941

compressor piston, more re-expansion, and less perfect valve action. In an electrically driven refrigerator, voltage increase may cause the energy consumption to go up or down, depending upon the design of the motor and the loading.

Specifications for testing mechanically operated refrigerators are set forth in the NEMA pamphlet #47-127 entitled "American Standard Test Procedures for Household Electric Refrigerators." **Figures 14** and **15** show performance of typical 8 and 10 cubic foot refrigerators in one manufacturers' line. **Figures 16** and **17** show a performance range of refrigerators produced by nine different manufacturers during 1949.

External and interior refrigerator temperatures may be measured by liquid in glass thermometers, electric thermocouples or electric resistance thermometers. **Glass thermometers** are relatively inexpensive but the refrigerator door must be open in order to read them within the refrigerator interior. While this may not affect the accuracy of the reading, it slows up the testing of the refrigerator as the box temperatures must be stabilized again before initial kilowatt hour readings or running time readings can be taken.

Resistance thermometers are accurate and reliable for measuring cabinet and room air temperatures but they are usually too bulky to measure surface temperatures on evaporators, tubes, compressors, and so forth. **Thermocouples** with potentiometer indicators are the most satisfactory means for measuring temperatures for development work. This is because thermocouples are cheap, extremely small, and can follow any desired rate of temperature change. They can also be lagged by putting them in contact with a mass of metal or fluid to read average rather than momentary cycling temperatures. Recorders are available for use with thermocouples or resistance thermometers that will plot as many as 16 records on one chart.

35. During testing it is important that the **temperature gradients** throughout the test room be kept to a minimum because of the effect of room temperature on refrigerator performance. The temperature at the floor should not be essentially different from that in the rest of the room, or the cold floor air being drawn through the condenser will give fictitious results. When testing a thermal-convection cooled condenser located on the back of the refrigerator, it is necessary to keep the room air velocity low enough so that heat dissipation from the condenser will not be affected. The NEMA Household Electric Refrigerator Standard requires that the room temperature be maintained within plus or minus 1 F of the specified value, and that the vertical temperature gradient from the floor to a height of 7 feet shall not exceed a half a degree for each foot of vertical distance.

The best method for making an **ice freezing** test is to measure ice freezing time by means of thermocouples inserted in the ice cube that will freeze last. As the water cools, the temperature in the trays will go down rapidly to 32 F. During the actual freezing period, the temperature will stay at 32 F and at the conclusion of this period, will go down rapidly from this point.

If an ice tray is placed on the shelf of a dry evaporator, there will be an entirely different rate of ice freezing than if the contact surfaces between the tray and the evaporator were thoroughly wet before the test. Differences as great as 2 to 1 in ice freezing time can result from this contact variation. Therefore, it is the usual procedure to wet the evaporator surface before putting the tray in place.

Kilowatt hours can be readily measured by an integrating kilowatt meter having a scale multiplier of .01 or less so that the data can be read to 1/100 kwh and estimated to 1/1000 kwh. The **running time** of the refrigerator unit can be measured by means of self starting electric clocks.

36. While there are no standard specifications for testing refrigerators under simulated **field operation,** such testing is desirable. An article, discussing this type of testing, appeared in *Refrigerating Engineering.*[3]

In setting up a laboratory test procedure that would simulate field conditions reasonably well, there are five variables that should be considered:

a. Dry bulb temperature
b. Dew point temperature
c. Number of refrigerator door openings.
d. Length of door openings
e. Usage load

To simulate severe field usage, the following conditions may serve as a pattern for a suitable test procedure. The test room condition is based on information contained in the work of J. C. Albright.[4] A **dry bulb** temperature of 90 F is suggested. This is 2 deg below the dry bulb temperature which is not exceeded more than 5% of the 12 hours during the middle of the day during June to September, inclusive, of a normal summer along the Gulf Coast. The 78 F **dew point** represents a condition along the Gulf Coast which was not exceeded more than 5% of the total hours during June to September, inclusive, for a normal summer. Roider[5] has suggested a slightly higher dew point of 81 F. Some data available indicates that not more than 2% of the refrigerator users **open the refrigerator door** more than 80 times per 24 hours. There is some deviation from this figure, depending upon the season of the year and the size of the refrigerator. Roider suggests a figure of 10 door openings per 24 hours per cubic foot of refrigerator volume. A **door opening period** of from 10 to 15 seconds is suggested. Laboratory tests indicated that at 60 F dew point and 90 F dry bulb, door opening periods of greater than 10 seconds duration have little effect on the amount of moisture pickup on the evaporator. Over a 12 hour interval, 80 door openings of approximately 13 seconds duration can be worked into a schedule of 6 approximately 1 hour periods of door openings. It is suggested that to duplicate the **food load** in the refrigerator, an equivalent weight of 78 F water in glass jars be substituted for the food in a fully loaded refrigerator. A schedule of twice a week shopping and the preparation of three meals a day can be assumed, and the simulated food added to or taken from the refrigerator accordingly. In addition, one tray of ice cubes should be frozen for each of the three meals.

37. Much can be learned from the standpoint of development by operating the compressor unit with a **calorimeter** of some kind. One type of calorimeter consists of a relatively short copper tube surrounding a tubular electric heater. Refrigerant is admitted in one end of the tube through a constant pressure expansion valve, and the heat input adjusted until the desired degree of superheat is obtained at the output. A modification of this particular idea has been used in connection with flooded evaporators. One feature of a good calorimeter is a small thermal mass so that its response to a change is rapid, thus reducing testing time. A calorimeter achieving these characteristics particularly well was designed by D. D. Wile.[8] It employs a heater immersed in the liquid of a secondary refrigerant circuit, while the evaporator of the compressor under test functions as a condenser for this secondary circuit. The control on the calorimeter turns the heater on and off in response to the pressure of the secondary system while the total heater input over a period of time is measured by a kilowatt hour meter. Accurate tests can be obtained from readings extending over a period of only three minutes.

38. Cabinet heat leakage tests can be run in several ways. (a) The cabinet may be placed in a hot room and the refrigerant passed to the evaporator at a rate necessary to maintain a constant cabinet temperature. The heat leakage per degree may then be determined from the quantity of refrigerant circulated, the condition of the refrigerant entering and leaving and the interior and exterior cabinet temperature. (b) The cabinet may be placed in a cool room (about 50 F) and the cabinet interior heated with electric heaters to a point where the mean temperature of the insulation is about 75 F. The overall heat loss may then be determined from the heater input and the interior and exterior cabinet temperatures. (c) The cabinet may be operated in a hot room with a calibrated refrigerating unit of the type normally installed in the cabinet. This method takes into account the localized heating of cabinet surfaces from the condenser and compressor, and gives the most accurate overall results under operating conditions approaching normal. (d) Ice may be placed in the cabinet and its rate of melting measured.

The Operating Cycle of the Heat-Operated Servel Refrigerator

39. The Servel refrigerator is heat-operated, the **freezing unit** being an absorption machine of the Platen-Munters three-fluid type. The unit is made up of a number of steel vessels and pipes welded together to form a hermetically sealed system. All spaces in the system are in open communication with each other and hence are at the same total pressure, aside from minor variations which arise from the presence of fluid columns and which are used to circulate the fluids.

The **charge** includes an aqua-ammonia solution of a strength of about 30% concentration (ammonia by weight) and hydrogen. For a unit of sufficient capacity for a 5-cu ft cabinet, the approximate charge is 1.1 lb ammonia, 2.6 lb water, 0.03 lb hydrogen. The liquid is charged into the unit as solution and then hydrogen is added.

The **elements** of the system in **Fig. 18** include: (1) a generator, sometimes called boiler or still; (2) a condenser; (3) an evaporator; (4) an absorber; and (5) a hydrogen reserve vessel. There are three distinct fluid circuits in the system: an ammonia circuit including the generator, condenser, evaporator and absorber; a hydrogen circuit including the evaporator and absorber; and a solution circuit including the generator and absorber.

40. Starting with the **generator,** heat is applied by a gas burner or other source of heat to expel ammonia from solution. The **ammonia vapor** thus generated flows to

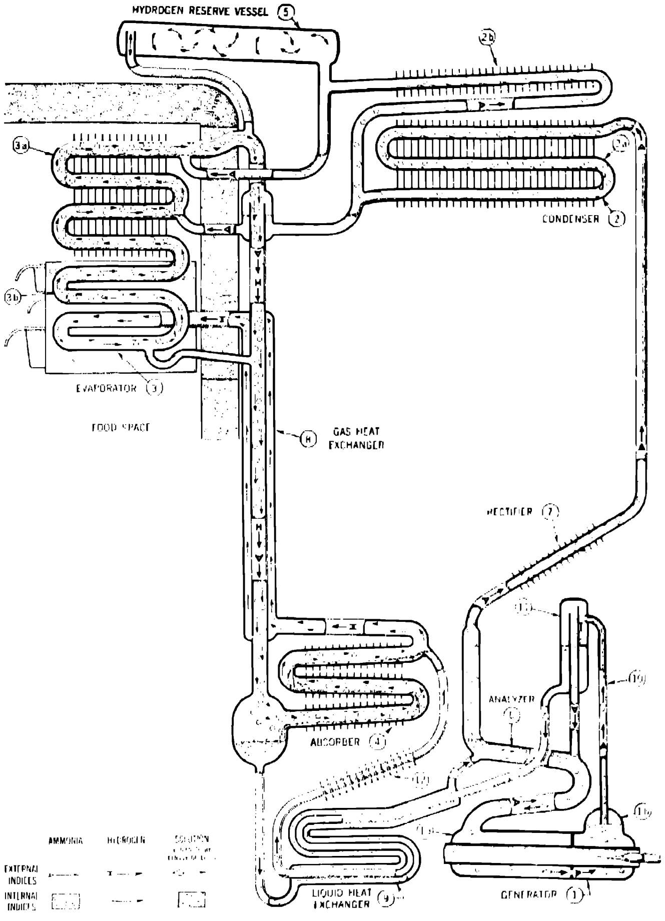

Fig. 18. Schematic Arrangement of Servel Absorption Machine (Platen-Munters System)

the condenser. In the path of flow of ammonia from the generator to the condenser are interposed an analyzer (6) and a rectifier (7). Some water vapor will be carried along with the ammonia vapor from the generator. The analyzer and rectifier serve to remove this water vapor from the ammonia vapor. In the analyzer, the ammonia passes through strong solution which is on its way from the absorber to the generator. This reduces the temperature of the generated vapor somewhat to condense water vapor and the resulting heating of the strong solution expels some ammonia vapor without additional heat input. The ammonia vapor then passes through the rectifier (7) where the residual small amount of water vapor is condensed by atmospheric cooling and drains to the generator (1) by way of the analyzer (6).

The ammonia vapor, which is still warm, passes on to the section (2a) of the condenser (2) where it is liquefied by air cooling. The condenser is provided with fins for this purpose. The ammonia thus liquefied flows into the evaporator (3) at an intermediate point. A liquid trap is interposed between the condenser section (2a) and the evaporator to prevent hydrogen from entering the condenser. Ammonia vapor which does not condense in the condenser section (2a) passes to the section

(2b) of the condenser and is liquefied, and flows through another trap into the top of the evaporator.

The **evaporator** is made up of two sections (3a and 3b). The upper section (3a) is provided with fins and directly cools the food space. The lower section (3b) is in direct contact with the ice-freezing compartment.

Hydrogen gas enters the lower evaporator section (3b) and, after passing through a precooling pipe part, flows upward, in counterflow to the downward flowing liquid ammonia. The effect of the placing of a hydrogen atmosphere above the liquid ammonia in the evaporator is to reduce the partial pressure of the ammonia vapor in accordance with Dalton's law of partial pressures. While the total or gage pressure in the evaporator and the pressure in the condenser are the same, there is substantially pure ammonia in the space where condensation is taking place, and consequently the vapor pressure of the ammonia substantially equals the total pressure.

Under Dalton's law, the total pressure of a gas mixture is equal to the sum of the partial pressures of the individual gases. Consequently, in the evaporator the partial ammonia vapor pressure is less than the total pressure by the value of partial pressure of the hydrogen. The lesser ammonia vapor pressure results in evaporation of the ammonia with consequent absorption of heat from the surroundings of the evaporator and the cooling of the surroundings which are in a well-insulated enclosure.

The cool heavy gas mixture of hydrogen and ammonia vapor formed in the evaporator leaves the top of the evaporator and passes downward through the center of the gas heat exchanger (8) to the absorber (4). In the **absorber,** ammonia is absorbed by water, and the hydrogen, which is practically insoluble, passes upward from the top of the absorber through the external chamber of the gas heat exchanger (8) into the evaporator. Perfect separation of gases is of course not possible and some ammonia vapor passes with the hydrogen from the absorber to the evaporator. It is probably more accurate to call the gas flowing from the evaporator to the absorber strong gas (hydrogen strong in ammonia), and the gas flowing from the absorber to the evaporator weak gas (hydrogen weak in ammonia).

Since the density of a gas is proportional to its molecular weight, and the molecular weight of ammonia is 17 and the molecular weight of hydrogen 2, it follows that the specific weight of the strong gas is greater than that of weak gas. This difference in specific weights is alone sufficient to initiate and maintain circulation between the evaporator and the absorber. Since the absorber is below the evaporator, it is possible to have upward gas flow in the evaporator. The long vertical column of strong gas in the central chamber of the gas heat exchanger is heavier than the vertical column in the absorber, external heat exchanger space and evaporator, despite the fact that the gas in the evaporator is heavy. Consequently the gas will flow as above stated, due to the difference in specific weights of the gases in the different vertical branches of the circuit. The gas heat exchanger transfers heat from the weak gas to the strong gas. This saves some cooling in the evaporator by precooling the entering gas. A liquid drain at the bottom of the evaporator is connected to the down-flow space of the gas heat exchanger.

Counter-current flow in the evaporator permits the location of the box cooling section of the evaporator in the most effective position, at the very top of the food space. Also, the gas leaving the lower temperature evaporator section (3b) can pick up more ammonia at the higher temperature prevailing in the box cooling evaporator section (3a), thereby increasing capacity and efficiency. There is still another advantage in that liquid ammonia flowing to the lower temperature evaporator section is precooled in the upper evaporator section.

The dual liquid connection between the condenser and the evaporator is advantageous in applying the unit to the cabinet. It permits extending the condenser below the top of the evaporator to provide more surface while having gravity flow of liquid ammonia to the evaporator.

The **two-temperature evaporator** partially segregates the ice-freezing function from the box-cooling function. This provides a better humidity condition in the food space because, due to the higher temperature of the box cooling section (though adequately low for proper preservation), and due to the reduced surface of the low temperature section permitted by the partial segregation of function, less moisture is extracted to form frost.

In the absorber, a flow of weak solution (water weak in ammonia) comes in direct contact with the strong gas. The liquid and gas flow is countercurrent. The weak solution is thus enriched or strengthened while the strong gas is weakened.

From the absorber the strong solution flows through the liquid heat exchanger (9) to the analyzer (6) and thence to the strong liquid chamber (1a) of the generator (1). Heat applied to this chamber causes vapor to pass upward through the analyzer (6) and to the condenser. The solution passes through an aperture in the generator partition into the weak liquid chamber (1b). Heat applied to this chamber causes vapor and liquid to pass upward through the small diameter pipe (10), as in an "air lift," to the separation vessel (11). Liberated ammonia vapor passes through the analyzer (6) and thence to the condenser. The weak solution flows through the liquid heat exchanger (9) and to the absorber. The liquid heat exchanger precools the liquid entering the absorber and preheats the liquid entering the generator. Further precooling of the weak solution is obtained in the finned air cooled loop (12) between the liquid heat exchanger and the absorber.

The heat which is liberated by absorption of ammonia in the absorber is carried away by air flowing in contact with the absorber fins.

The **hydrogen reserve vessel** (5) which is connected between the condenser outlet and the hydrogen circuit may be described as a reservoir for hydrogen gas while the refrigerator is operating under normal room temperature conditions. Under these conditions an appreciable part of the hydrogen in the system is stored in the reserve vessel. The remainder is located in the evaporator-absorber circuit and serves to balance the condenser pressure. The pressure must be adequate to liquefy the ammonia gas in the condenser. If the pressure is increased, the efficiency under normal conditions will be impaired, and yet it is necessary to have a higher pressure in the system to insure condensation of the ammonia under high room temperature conditions. The reserve vessel, through its connection in the system, is an automatic pressure variant to take care of the variable room temperature and loads. It permits lower operating pressure at lower room temperature, thereby resulting in better efficiency, but provides the higher operating pressure necessary under extreme conditions to insure condensation of ammonia.

HOUSEHOLD FREEZERS

41. Although the frozen food industry began a rapid growth about 1920, commercial production of refrigerated cabinets designed specifically for home and farm frozen foods was not seriously considered until about 1937. By 1948 industry shipments of freezers had reached 690,000 units.[7] While the food shortages during the war greatly favored the growth of this product, the subsequent years have seen a growing public acceptance of this product. **Table 4** shows a breakdown of the 1949 sales of 31 National Electrical Manufacturers Association companies. An examination of the sales since 1947 by sizes of cabinets indicates a shift in the highest volume of sales from the 6 to 8 cu ft sizes to the 12 to 13 cu ft sizes.

42. The functions of the household freezer are to provide a storage space at a temperature of 0 F or below and a fast freezing zone where limited quantities of fresh food may be frozen in a reasonably short time with minimum disturbance of the stored load temperature. These functions should be performed with the following objectives in view:

a. Minimum temperature variations due to cycling of the refrigerating unit
b. Well insulated cabinet for economical power consumption
c. Minimum external surface temperatures in line with household refrigerator practice
d. Ease of access to all parts of refrigerated space
e. Efficient, reliable and quiet refrigeration system

Table 4. Home and Farm Freezer Sales Statistics for National Electrical Manufacturers Association Companies, for year 1949[8]

Sizes	Domestic (48 states and D.C.)	Canadian	Other foreign	Total
1. 4.9 cu ft and under	22,126	—	1,296	23,422
2. 5.0 to 6.9 cu ft	38,725	2	904	39,629
3. 7.0 to 8.9 cu ft	73,977	1	2,045	76,023
4. 9.0 to 10.9 cu ft	22,919	—	91	23,010
5. 11.0 to 12.9 cu ft	69,186	—	197	69,383
6. 13.0 to 16.9 cu ft	49,881	—	119	50,000
7. 17.0 to 20.9 cu ft	28,223	—	69	28,292
8. 21.0 to 29.9 cu ft	1,648	—	6	1,654
9. 30.0 to 39.9 cu ft	2,468	—	—	2,468
10. 40.0 to 49.9 cu ft	20	—	—	20
11. 50.0 to 59.9 cu ft	1	—	—	1
12. 60.0 cu ft and over	53	—	—	53
Total All Models	309,227	3	4,727	313,955
13. Total Upright Models (Included in above)	13,721	2	62	13,785

f. Sturdy construction including adequate moisture vapor sealing

g. Durable and attractive appearance with low odor levels considered in the choice of internal materials.

Cabinets

43. Mass produced cabinets are predominantly of the chest type with all-steel construction and synthetic enameled exteriors. The exterior shell is stiffened at the base by the bottom pan, which supports the insulation, and a sub-base or angle iron frame. The top of the shell is strengthened by a rigid tie with the interior compartment. This rigid tie may be in the form of a wood frame with a decorative breaker strip of rubber or plastic, or plastic laminates may be used for both the decorative and structural functions. Aluminum has received preferences for the construction of interior compartments with baked enamel on steel and aluminum paint on galvanized steel also in general use. Interior compartments are generally supported by the bottom pan of the outer shell on blocks of rigid insulation. The recent trend in lid design has been away from the plug type and toward a single full-opening lid up to the 16 cu ft size. Lid construction generally consists of an inwardly flanged sheet steel exterior carrying a plastic laminate inner panel and a balloon-type rubber gasket. Counterbalancing springs and a lid latch with provisions for locking are standard equipment.

44. Upright freezers have followed the pattern of household refrigerators or the larger reach-in refrigerators. Interior locker doors on the face of the food compartment behind the main door are used to reduce air spillage when the main door is opened and may be arranged to form convenient loading shelves when open. The question of air spillage from upright cabinets has received considerable attention. Laboratory tests comparing chest and upright types indicate no significant difference in operating costs with a door opening frequency of 2 per day.[9] When the opening frequency is high, as in a commercial dispensing cabinet, the frost accumulation in the upright type may become objectionable, but in household use this objection has not been experienced.

45. Since a household freezer may be operated in damp locations such as a cellar or may be used in a modern kitchen, it is important that exterior surface temperatures be kept high enough to avoid objectionable condensation. The surface adjacent to the breaker strips is usually the critical area in this respect. A minimum cross-sectional area of materials of low thermal conductivity consistent with durability must be used. Air circulation across the surface of the breaker strips should be retarded as much as possible.

This should be accomplished by restriction rather than isolation of the air as the latter course will lead to a troublesome frost condition.[10] Surface temperature depressions of 4 to 5 degrees below ambient in a 70 F room should be acceptable. To insure freedom from condensation at this area, some manufacturers have used a low wattage insulated heater wire attached to the inner shell surface near the top. Center mullions in multiple lid cabinets are frequently a source of condensation. It is advisable to provide for free air circulation under the cabinet to avoid condensation under the bottom pan.

46. Insulation thickness varies widely but figures on 1949 freezers indicate that approximately 4 inches is used on the average except for the lid where considerably less insulation is usually provided.[10] Insulation has been predominantly of the glass fibre and mineral wool types in batt form. The so-called high density form of these insulations which offers slightly better insulation than standard grades has been extensively used. Recent changes in manufacture of these fibrous insulations have produced finer fibers with more insulating value per dollar of cost. Silica aerogel, certain expanded plastics and special gases replacing the air in mass insulation offer interesting possibilities for increasing the storage space by reducing the insulation volume with materials of exceptionally high insulating value.[11,12] Since a substantial portion of the heat leakage of a cabinet occurs through the gaskets, breaker strips and mechanical supports, it is difficult to realize completely the gain in insulating effect indicated by the low conductivity factors of these materials.

47. A high resistance to moisture vapor penetration through the outer shell of a freezer cabinet is of the utmost importance for satisfactory performance during the long operating life normally expected. While the accumulation of moisture in the insulation by the process of diffusion can be controlled by mediocre sealing of the shell, the influx of moisture laden air due to the small changes in internal air pressure which occur with cycling of the refrigerating unit (and the less frequent pressure changes due to door openings) is independent of the area of the opening. In actual practice the pressure difference may be continually equalized by some venting arrangement such as the lid or door gasket, but the design must be such that the moisture entering such a vent will be deposited on a surface where it can be readily removed during defrosting. The process will work effectively only if the shell is sealed to a point where its resistance to air flow is much greater than the resistance of the venting means.[13] Joints in the external shell are sealed with various forms of wax, tars, and plastisols. The effectiveness of sealing has been effectively checked by both air pressure and vacuum methods.[14]

48. The question of condensing unit location has received three treatments. The small sizes of cabinets generally have a shallow food compartment which allows the condensing unit to be located under the bottom insulation. This arrangement offers a more accessible interior than the deeper compartments, but the ratio of floor area to usable volume is high. The lowest ratio of floor area to usable volume is offered by the design which locates the compressor under a step in the food compartment. This arrangement, which may require more costly fabrication, offers a convenient fast freezing zone if the bottom as well as the vertical surfaces of the shallow portion of the compartment is refrigerated. In the larger cabinets the condensing unit may be contained in an enclosure added to one end of the insulated cabinet. This design usually results in waste volume above the condensing unit, and in some cases, this has been arranged as a food thawing zone or a storage compartment for wrapping materials.

49. Equipment for the convenience of the user which may be factory installed or sold as accessories include such items as: wire shelves, baskets and partitions, interior light, temperature indicator, warning device, and insurance policies covering the contents of the freezer against loss by certain types of refrigeration failure.

Refrigerating Systems

50. The evaporator in the majority of chest type freezers today consists of tubing bonded to the exterior surface of the food compartment. Generally the tubing is placed only on the vertical walls except where it is desired to form a fast freezing surface by refrigerating a portion of the bottom or a shelf. A common form of this type of evaporator uses copper tubing which is held against the exterior of the compartment wall by various types of clips or clamps. A mastic with good thermal conduction may be used as a filleting material. Aluminum food compartments and aluminum tubing may be joined in an intimate mechanical and thermal bond by the use of a special brazing process. The expensive equipment required for this process as well as subsequent surface treatment of interior surfaces has limited its use. The use of aluminum evaporator tubing introduces a special problem where connections are made to tubes other than aluminum. This problem may be solved by the use of special pre-welded joints which are brazed in place by the customary processes. Wide variations are found in the choice of tube spacing and length. The factors of cost and pressure drop in the evaporator tubing dictate a short tube length on necessarily large centers, whereas the factors of temperature drop from tube to wall and in the wall between tubes call for a long tube length and small center distances. Chest type freezer evaporators are also formed from refrigerated plates mounted vertically, parallel to the food compartment walls, or as dividers between sections. These plates may be of tube and plate type, or the brazed double wall type with integral refrigerant passages. Upright freezer evaporators usually consist of horizontal refrigerated plates which act as shelves. Another evaporator system consists of a plate surface cooled by the primary refrigerant in the upper part of the cabinet and a coil surrounding the food compartment which receives its refrigeration from a secondary refrigerant in heat exchange relation with the primary system. (Fig. 19.) This system is attractive from the standpoint of offering a food compartment free of frost, a cold plate for fast freezing and localization of frost and the sealed, removable refrigerating unit which has proved so successful in household refrigerators. However, the cost of such a system is relatively high, and a careful design is required to avoid excessively low evaporating temperatures with resultant loss in refrigerating unit capacity.

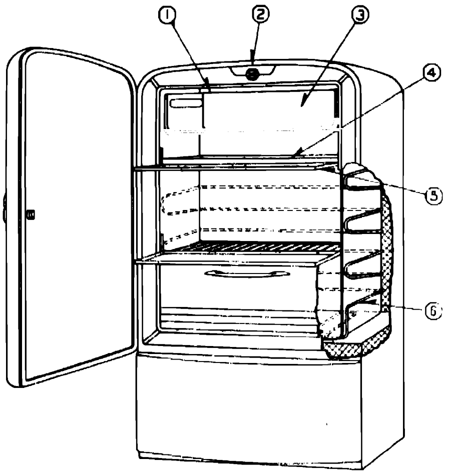

Fig. 19. 6-cu ft Upright Home Freezer

1. Evaporator with secondary system condenser attached at back
2. Temperature adjusting knob
3. Freezing compartment
4. Defrost tray and shelf
5. Compartment doors
6. Secondary system evaporator coils on sides, back and bottom of compartment

Condensing Units

51. Experience in the household refrigerator industry has indicated the use of hermetically sealed systems with capillary tube expansion device for freezers where reliability is of paramount importance[9] **(Fig. 20).** Ease of replacement should be considered.

52. The low suction pressures involved in this application require good compressor construction with high volumetric efficiency. Compressor motor loads will be relatively light in normal operation but sufficient power must be available for the initial pull-down of a warm cabinet or overloading due to excessive freezing loads. The selection of compressor lubricating oil involves the consideration of wax precipitation at the low temperatures and special oils which will meet this requirement should be used.[15,16,17] Permanent dehydrators in addition to thorough factory dehydration are frequently included in the refrigerant circuit as the moisture limit in these low temperature systems is extremely low.[18]

53. Condenser design has followed closely the household refrigerator practice in both static and forced draft types with some manufacturers using all or a portion of the cabinet shell as a plate condenser. As in compressor motor design the normal condenser load will be relatively small, but adequate capacity for peak loads must be provided.[17]

54. Freon-12 refrigerant is commonly used in this application. Freon-22 refrigerant has been used to some extent. The increase in refrigerating effect of this refrigerant over Freon-12 for the same displacement compressor sometimes offers economic possibilities by using an available mass produced compressor to operate a larger freezer or to operate a freezer in place of a refrigerator. Such a change should involve a reconsideration of the design of the condenser, expansion device, motor characteristics and evaporator with regard to the higher pressures and increased quantity of heat to be handled, and the difference in interaction with the compressor oil between the two refrigerants.[17,19,20]

55. While capillary tubes are usually used on the household sizes of freezers,[21] thermostatic expansion valves may be used, but particular attention is required, because of the small pressure change available for operating the valve at the low temperature range in this application.[22,23]

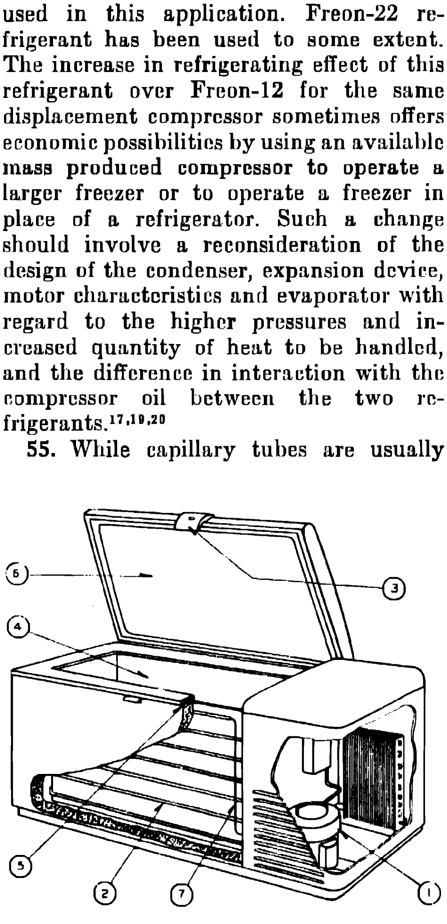

Fig. 20. 16-cu ft Chest Type Freezer

1. Condensing unit
2. Wall cooling coils
3. Locking lid handle
4. Storage compartment
5. Breaker strip construction
6. Plastic laminate inner lid pan
7. Capillary tube expansion device

56. Temperature controls for operating the refrigerating unit of household freezers are standard devices designed to operate at the required temperatures. The location of the temperature sensitive element requires careful consideration. A constant temperature of the stored load is highly desirable from the standpoint of preserving quality products for long periods of storage.[24,25,26] Temperature variations due to cycling of the refrigerating unit as well as those due to changes in ambient temperature should be held to a minimum. If possible, the temperature control should automatically call for continuous operation of the refrigerating unit until fresh loads of food are frozen.

Storage and Freezing Temperatures

57. Extensive research in the field of the quality of frozen foods versus storage time and temperature has established a temperature of 0 F as satisfactory for most foods for a storage period of six months to a year.[25,26,27] Certain foods, particularly pork and its products, require either lower temperatures or shorter storage periods.[28,29]

58. The effect of the rate of freezing on the quality of frozen foods has been extensively explored. Early authorities have, perhaps, overstressed the necessity of quick freezing. A summary of the literature leads to the conclusion that high quality can be achieved with relatively slow freezing providing that recommendations regarding the other factors in this process of food preservation are carefully followed.[25] In the design of the household sizes of freezers provisions should be made for the freezing of limited quantities of food in proportion to the size of the freezer, and specific instructions should be available to the user as to the proper placement of food to be frozen and the maximum quantity that may be frozen in 24 hours without seriously disturbing the stored load temperature. The maximum rise in storage temperature has been considered as

5 degrees.[30] On the assumption that instructions are followed, the provision of such freezing facilities will insure the rapid cooling of the warm load to a point where spoilage is at a minimum. Freezing by contact with horizontal refrigerated plates or by rapidly moving air offer the best possibilities for quick freezing in the household freezer.[31] Adequate condensing unit capacity will be required to carry the recommended freezing load in addition to the normal operating load.

Refrigeration Failure

59. Since the financial loss through spoilage of the contents of a freezer can be of serious magnitude the problem of refrigeration failure has received considerable attention.[27,32] The effects of ambient temperature, quantity of stored load, door openings, and individual cabinet heat leakage seriously affect any estimate of the time available between loss of refrigeration and spoilage temperatures. In general, thawing of at least part of the food may be expected within 30 hours and temperatures exceeding a safe limit of 50 F may occur in 75 hours. The application of dry ice is recommended as a temporary source of refrigeration.

Early knowledge of the failure of refrigeration is of primary importance in avoiding food spoilage. A variety of warning devices and temperature indicators are available. These devices are offered by many freezer manufacturers as standard equipment or as an accessory. To obtain the maximum protection the thermal element of such a device must be carefully located with regard to temperature distribution within the freezer.[33]

Food Processing and Packaging

60. Regardless of the excellence of the freezer design and manufacture with regard to the points covered in this article the user of the freezer must be made fully aware of his responsibilities in its use and in the fundamental steps of food processing and packaging in order to achieve a high quality product.[34,35,36] Only the essentials of this process can be mentioned here. These consist of: a high quality fresh product of proper maturity and recommended variety, the shortest possible time between picking and freezing of certain types of vegetables and fruits, uniform scalding of vegetables plus immediate cooling, cleanliness, and packaging in moisture vapor proof material with the maximum possible exclusion of air from the package.

Testing and Rating

61. The testing and rating of household freezers is covered by National Electrical Manufacturers Association Publication 45-106, quoted in part below.

HR9-5. Rating the Storage Capacity for Home Freezers

A. Food Storage Volume Rating
  1. The storage volume of home freezers shall be the volume in cubic feet of the space specifically arranged for the storage of frozen foods, except as modified in paragraphs 2 and 3.
  2. The volume of any space arranged for freezing shall be included in the storage volume since this space may also be used for storage.
  3. In determining the storage volume, the volume of permanently fixed shelves and partitions shall be deducted from the gross volume.

B. Food Storage Weight Rating

The storage volume rating may be supplemented by a food storage rating which shall be 35 pounds per cubic foot of storage volume. (See Note 1.)

Example. Storage volume 6 cu ft
Food storage capacity, 210 lb

Note 1.—Approximately 40 lb per cu ft loading is obtained with uniform size packages of chopped meat or fruits in syrup. Approximately 30 to 35 lb per cu ft loading is obtained with miscellaneous meats in family-size cuts. Approximately 25 to 30 lb per cu ft loading is obtained with an assortment of packages, including fruits, vegetables and miscellaneous cuts of meat.

Note 2.—It is recommended that the manufacturer include in his user instructions the maximum number of pounds of fresh foods which should be frozen in any 24 hr period.

Codes and Specifications

1. American Standards B38.1-1944. Method of Computing Food Storage Volume and Shelf Area of Automatic Household Refrigerators.
2. American Standards B38.2-1944. Test Procedures for Household Electric Refrigerators Mechanically Operated.
3. American Standards B9.1-1950. Safety Code for Mechanical Refrigeration.
4. Underwriters Laboratory Standard for Unit Refrigerating Systems, dated Feb. 1944.
5. National Electrical Manufacturing Association 1950. General Specifications for Household Refrigerators.

6. American Standards Z21-19, 1936. Approval Requirements for Refrigerators Using Gas Fuel.

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30. INSULATION OF CABINETS

A MODERN refrigerator cabinet must be designed and constructed so that the heat flow rate from the surrounding room into the interior space is reduced to a point where a refrigerating unit reasonable in initial cost, operating cost, and practical in regard to weight and size can maintain consistently satisfactory refrigeration temperatures.

In addition, a sufficiently large temperature differential between inside air and outside surface must be introduced to maintain the outer surface at a temperature close enough to the ambient that undesirable **water condensation** on the exterior does not occur.

Cabinet design features and details, type and thickness of insulation selected and used, and the quality and uniformity of manufacturing operations all effect the overall heat leakage of the completed refrigerator cabinet.

Cabinet Design Features

In addition to the heat flow through the insulated walls an appreciable portion of the total cabinet heat leakage is a result of heat flow through structural members, edge construction, door structure, latch mechanism, faulty gasket, seals, etc.

To minimize conduction through structural parts and particularly the conduction termed edge losses, structural and trim members should be as small in size as is consistent with good mechanical design, and should be made of low thermal conductivity material possessing the characteristics required. Highly conductive "bridges" such as unfilled voids, unbroken metallic connections, or insulation displacing functional parts or devices often become heat-flow paths that greatly increase the localized conductivity. Surface condensation troubles most frequently arise in areas where such highly conductive bridges occur. Latch and hinge areas are particularly apt to prove troublesome in regard to objectionable surface condensation. In **Figure 1,** if the dew point of the room air lies betweem the temperatures of points *X* and *Y*, sweating will occur on the refrigerator surface at *Y*.

The extra-insulation cabinet heat leakage can be minimized through the:

a. Careful avoidance of direct metallic connection between interior and exterior parts.

b. Use in both the door and cabinet

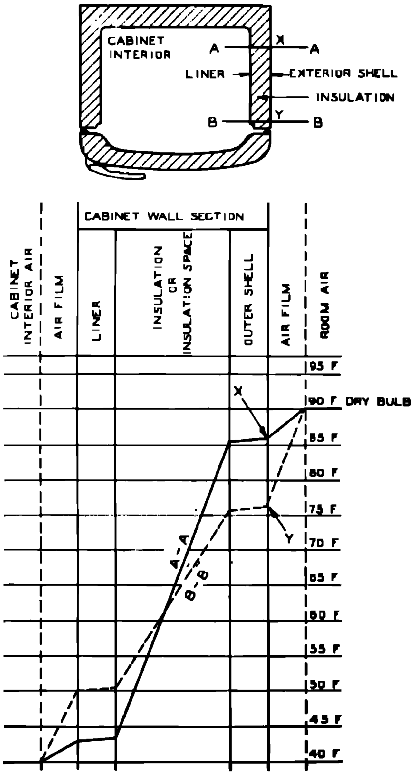

Fig. 1. Temperature Gradient

(Shows that point *Y* on exterior shell of high conductance section *B-B* is at a considerably lower temperature than point *X* on exterior shell of low conductance section *A-A*.)

proper of present day available plastics for both major and minor structural members—particularly for bridging between metallic structural parts.

c. Use of plastics for breaker-strips or other functional and decorative trim parts.

d. Use of resilient rubber or other elastomer door gaskets plus effective door latching mechanisms that combine to provide non-leaking door seals.

e. Careful similar attention to leakage at all other necessary openings through the insulated walls such as cooling unit, connecting tubing and other entries.

In some cases design features or details at certain regions, such as the dividing stile or transom in multi-door cabinets, can at best not result in surface temperatures high enough to avoid objectionable surface condensation. In such cases electrical heat is frequently added to further elevate the surface temperature.

Selection of Insulation

The characteristics of an insulating material that are of major importance in the selection of a suitable material for refrigerator cabinet use are: conductivity, overall cost, permanence, and low odor level. Other characteristics of varying degrees of importance are: density, moisture resistance and other moisture-related properties, health hazards, flammability, and insect and vermin repellent qualities.

As the decision regarding the thickness of insulation most suited for use in a specific refrigerator cabinet involves consideration of several of these characteristics, we will first discuss these, and then treat the subject of thickness selection.

a. The **conductivity** of the insulation is expressed as *k*, expressed in Btu in. per sq ft hr F temperature differential. Current refrigerator cabinet practice uses insulations with *k* factors in the range of .24 to .28. However, insulations are available and may be used with conductivities ranging from approximately .145 (opacified silica aerogel) to .36 (semi-rigid cellulosic building board).
The relationship of insulation *k* factor to heat transmission rate through the refrigerator wall is discussed later under "Cabinet Heat Transmission Rate."

b. The **overall insulation cost** includes numerous factors:
    1. **Basic cost of the insulation.** This may be the approximate average of about 1.8 cents per board foot for the insulations commonly used or may run as high as 15 cents or more for some of the less commonly used materials.
    2. **Cost at point of application.** This cost reflects the availability of the material, shipping charges to place of use, handling and storage costs, and cost of preparation. Some insulating materials are so fragile that special and costly handling and storing facilities must be provided.
    3. The overall-cost of any insulation study must include the **cost of installation** in the cabinet. Non-fragile materials that are quickly and satisfactorily installed in the cabinet under construction obviously have a cost advantage over materials requiring special handling, frequent repairs or patches, or the payment of bonus labor rates due to skin irritability or other objectionable features.
    Some insulating materials may be blown into the insulation spaces and direct hand application thus avoided. However, the use of easily handled materials, the correct dimensioning of insulation pads, the uniform maintenance of sizes by the supplier, well designed cabinets, and well planned assembly procedures have combined to make possible very low application costs.

c. The **permanence** or life of the insulation depends upon the chemical and mechanical stability of both the raw materials and the finished insulation. Freedom from shrinkage due to loss of water or any other constituent and freedom from settling caused by jarring and vibration during manufacture, crating, handling, shipping, installing and either normal or abnormal use are of major importance. Only insulating materials that pass the most rigid jolting, dropping, and vibration tests can be considered for modern refrigerators.

d. The insulating material itself as well as all materials which may be used in conjunction with it such as binder, sealer, wrapper, or container must be low in **odor and taste level,** unless the specific usage so seals the insulation that odors are positively retained and prevented from contaminating the food space or its contents. In addition to not possessing an odor of its own a truly satisfactory insulation should also not pick up and retain odors to which it may be accidentally subjected.

e. The **density** of the insulation is actually a factor in the overall cost. Heavy insulations increase total costs through higher transportation, handling and storage costs for both the insulating material and the completed refrigerator.

For mobile equipment the density becomes of greater importance because high density insulations decrease the ratio of pay-load to gross weight. As a result, the density factor combined with the k factor may be of greater importance than the basic cost of the insulation.

f. **Moisture Resistance—Cabinet Sealing.** This characteristic was omitted from the listing of major properties because present day good design and manufacturing practice call for practically vapor tight barriers that prevent the entrance of moisture laden air into the insulation space and the subsequent deposition of water at all points lower in temperature than the dew point of the entering air. By thus eliminating water in contact with the insulation the properties of moisture resistance, capillarity and effect of moisture on the k factor become of minor importance. As only a few insulating materials do not deteriorate when wetted, the importance of maintaining dry insulation cannot be over emphasized. In most modern day cabinets the vapor seal required is effected by designing the cabinet structure of sheet steel with a minimum of joints and sealing all unavoidable joints with effective and permanent sealing compounds. Asphalt, waxes, and compounded mastics are commonly employed.

As a truly hermetically sealed cabinet is not practical or attainable in all cases in large volume production, common refrigerator design for some types calls for accidental or even intentional openings from the insulation space to the low humidity refrigerator interior. However the advent of refrigerator types with low temperature refrigerated coils within the insulation space has necessitated even closer attention to the obtaining of a good warm-side vapor barrier and has even directed some attention to vapor seals to the cabinet interior. This is especially true in designs in which a single insulation space encloses several compartments at varying temperatures and humidities. Easy access of air from a high temperature high humidity compartment may result in diastrous water deposition around a low temperature compartment.

g. **Freedom from Health Hazards.** Vapor, dust, loose particles, etc. arising from the insulation during cutting, handling, moving, applying, etc. must not introduce definite health hazards or even be sufficiently objectionable to cause serious personnel problems. Minor personnel problems arising from objectionable but not seriously hazardous pulmonary dust and skin irritation cases from otherwise very desirable insulations can usually be solved by selecting personnel of reduced specific sensitivity and in some cases by a pay rate bonus.

h. **Flammability** is of minor importance after assembly into the cabinet, but may be of some importance relative to handling and storage.

i. **Insect and Vermin Repellent Qualities.** In currently produced well sealed refrigerators this property is of minor importance. However for universal use insulating materials that are repellent as food or home for insects and vermin have an advantage.

Types of Insulating Materials

Many different materials and types of materials have been used and considered for use in refrigerator cabinets. It is not within the scope of this chapter to discuss at length the many possible materials. For reference to further information and discussion of the various insulating materials see References 1 through 15 in the appended bibliography. Frequent classifications have been made as to types, for example: organic, inorganic, cellulosic, fibrous, rigid, semi-rigid, bulk, loose-pack, derived from plastics, powder, natural air-cell, air-cell fabricated, cellular, reflective, vacuum, filled-vacuum, gas-filled cell, sub-ultra microscopic air cell, and others. Currently, the insulation used in the majority of refrigerator cabinets is the inorganic type—mineral, rock, or glass fiber, either nonrigid felted, or formed. Application is by sized and pre-cut slabs or batts.

Thickness of Insulation

The design thickness of insulation, other cabinet design details, and the *k* factor of the insulation selected, are important factors in determining the **overall heat leakage** rate of the refrigerator cabinet. In the determination of the thickness to be used a compromise must be effected between several related factors. These include:

a. The relationship of *k* factor and cost. In some instances an insulation with a *k* factor of .26 and a cost of 1.8 cents per board foot may be the most economical, whereas in other cases an insulation with a *k* factor of .22 and a cost of 3.2 cents per board foot may be dictated.

b. The relationship of *k* factor, insulation cost and other numerous possible manufacturing costs. To avoid the cost of tooling for a differently sized cabinet an 8 cu ft refrigerator might advisedly be turned into a 9 by the use of higher cost but lower *k* factor insulation.

c. The evaluation made and emphasis placed on the ratio of usable internal volume to bulk space occupied. For "cramped-quarter" usage such as apartment kitchens, trailers, yachts, the use of high priced low *k* insulation may be advisable, purely for the reduction in refrigerator size.

d. The effect of an unequal temperature gradient through the several walls of a refrigerator resulting from design or use factors—for example, the effect of a warm refrigerating unit or part of a unit below or in back of the cabinet.

e. The external surface temperature depression considered as acceptable without encountering objectional sweating. Manufacturers have various criteria as to what is an acceptable value. For example:

   1. One manufacturer may aim for a limit of an 8 deg F temperature difference when operating in a 100 F ambient.
   2. Another may use a rule-of-thumb requiring $1\frac{1}{2}$ in. of insulation for 40 F inside temperature and $2\frac{1}{2}$ in. for 0 F.
   3. A third may aim for a slight condensation pattern at 85% rh in a 90 F ambient.

In some localities at some hours wet bulb temperatures are within a few degrees of dry bulb temperatures. Designing so that absolutely no surface condensation results under such conditions is impractical. Design conditions are usually considered acceptable that keep condensation to a minimum, and prevent it for sustained periods which would result in water trickle downward over the surface.

Cabinet Heat Transmission Rate

The heat transmission rate for the cabinet is determined by the previously discussed items of: cabinet design features, *k* factor of insulation selected, thickness of insulation used. The rate of transmission may be calculated thus:

a. Wall insulated with one material

$$Q = \frac{kA(t_1 - t_2)}{L}$$

where: Q = heat transmission rate in Btu per hr
k = conductivity of insulation, Btu in. per sq ft hr F
A = mean wall area, sq ft
t_1 = temperature of refrigerator outer surface (warm side), F
t_2 = temperature of refrigerator interior surface (cold side), F
L = average wall thickness, in.

b. Wall insulated with several materials in series having thicknesses of L_1, L_2, L_3, etc. and respective conductivities of k_1, k_2, k_3, etc.

$$Q = UA(t_1 - t_2)$$

The value U is a heat conductance value for the composite wall section in Btu per sq ft hr F and is determined from:

$$\frac{1}{U} = \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \text{etc.}$$

A cabinet heat transmission value so calculated may be in error in several respects.

a. Edge losses, door losses, etc. are neglected, and these increase the actual transmission value to above the calculated value.
b. The selected values of $t_1 - t_2$ are usually not the correct values for the insulation itself. If, t_1 and t_2 are respectively the outside and inside air temperatures, the difference of $t_1 - t_2$, may exceed the true Δt across the insulation 6 to 12 deg F. Since the Δt used in the calculation exceeds the true Δt, the calculated heat transmission rate will exceed the actual value.

It has been noted that for some cabinets of current conventional design and construction the error of (a) above approximately equals the error of (b) when air temperatures are used for t_1 and t_2, and therefore, the calculated value approximates the test value.

Several test procedures[17] for determining a cabinet heat transmission value are in use. Frequently a measured electrical heat load is added to the cabinet of a non-operating refrigerator placed in 50 F room. The heat load is varied to a value that results in inside temperatures approximately 60 F above the test room temperature. Conditions are thus reversed from usual operating conditions. Measurements of the electrical heater input and the resulting actual Δt established easily permit calculation of a heat transmission factor for the specific cabinet on test. Such test results or cabinet evaluations may be expressed in units of Btu per hr for a one degree temperature differential or in lb IME (ice melting equivalent) per 24 hr for a 60 deg F Δt. (One lb IME equals 144 Btu.)

Factory Production Factors

a. **Shipping, storage, and handling**—The use of pre-cut bats of low density insulation, while very common, requires large inventories, and subsequently large storage space and related handling facilities. This is especially true in large volume producing operations in which several different sized insulation pads are required for each of several different sized cabinets. Shipment of bulk insulations in a high density state and then reducing to a usable low density state is one approach to lower shipping and storage costs. A half-way measure currently in use is shipping and storage in a somewhat compressed state. This method is obviously applicable only with insulations that have enough resilience to recover sufficiently from the compressed state to regain the density and physical properties required.
b. **Quality control**—The best cabinet and insulation designs and specifications may be entirely negated by faulty production line operations. Detection of insulation defects in assembled cabinets is difficult. To minimize the frequency of such defects, cabinet and insulation designs that permit rapid, simple and easy assembly of the insulation are particulary valuable. Other factors that aid in obtaining a uniformly satisfactorily insulated cabinet are: Selection of responsible labor personnel, adequate supervision and periodic or frequent checks by inspection personnel.

Refrigerator Performance Tests

The adequacy of the insulation design

and the compliance with this design are obviously factors entering into the overall refrigerator performance. Consequently performance tests, which should be conducted in accordance with ASA Standard B 38.2[18] are valuable to a large degree in determining whether or not the refrigerator cabinets are satisfactorily insulated.

A valuable adjunct to the ASA B 38.2 tests is exploratory surface temperature measurement with fine-wire and small-mass "touch" thermocouples. A satisfactory approximation of the exact surface temperature depression below ambient, as discussed in Par. (e) under "Thickness of Insulation," can be determined in this manner.

More exact test information in regard to surface temperature conditions can be determined by gradually raising the humidity in the performance test room and observing:

a. The areas at which condensation first occurs.
b. The location and extent of the sweating areas at some specific temperature and relative humidity conditions.

As good "long term" performance may be distinguished from "as new" performance, and as the continued performance over a period is largely dependent upon cabinet seal, methods of evaluating the cabinet seal will be described further.

Seal Tests

a. Vacuum tests[19]—A quickly applied test, that may be conducted on the refrigerator either after or before the door assembly, provided a temporary closure is provided, is by removing air from the insulation space with a vacuum pump until a pre-selected vacuum reading is obtained. The rate of return to atmospheric pressure shows the effectiveness of the cabinet seal. Semi-automatic well designed test stations have been put into operation that make it feasible to test cabinets on a production assembly line. In such setups, electrically operated timing and indicating devices show whether or not cabinets meet the pre-selected standard that is regarded as an adequate air-tight seal.

b. Air pressure tests[20]—In some instances a low value positive air pressure is applied to the insulation space instead of the vacuum as described. In such tests the rate of loss of pressure indicates the quality of the seal. The air pressure test setup can be made semiautomatic to the same degree as the vacuum test apparatus. However, in some instances in order to avoid damaging sealed joints by expansion, as the air pressure is applied, reinforcing fixtures must be employed.

   Both the vacuum and air pressure tests determine the size of seal leaks present but do not show the location of the leaks. A leak locating method that can be easily used to supplement the pressure type test is the common soap-water-bubble test.

c. Vapor tests—The combination of F-12 vapor introduced with slight positive pressure and a sensitive halide leak detector, preferably of the electrical type, affords a method of detecting the presence as well as the location of seal-leaks.

   Another extremely sensitive test and probably the one which most accurately determines the exact location of the leaks is the use of ammonia vapor under slight positive pressure in the insulation space with the hand application of wetted phenolphthalein paper over all joints or areas where leaks are possible. The slightest leakage of ammonia results in a pink or red stain on the test paper. These vapor tests are not quick tests as compared with the vacuum or air pressure tests, and are applicable only as laboratory tests or small percentage production tests. More reliable and more quickly obtained results are obtained with the vapor tests if conducted without insulation in the insulation space. Consequently, if so conducted, the application of vapor tight closures to all openings, such as door and unit openings, further complicates the test procedure.

d. Humidity room performance—Direct tests of the efficacy of the cabinet seal in keeping the insulation dry are conducted by further adjuncts to the ASA B 38.2 tests in which the refrigerator is operated over an extended period in a high humidity room. Conditions of 100 F and 90 % rh or 90 F and 85% rh are often maintained. Test periods of 30 days with a disassembly and analysis for water accumulation within the insulation are sometimes employed. However, tests extending over a three months' period produce more reliable results.

Some of the main objectives of cabinet insulation, and cabinet seal required for a long maximum efficiency life for insulation, are the minimization of energy consumption and the maintenance of that energy input level. Consequently, the most direct evaluation of the degree to which these objectives have been attained as obtained by a humidity room operating tests. In such a test, energy consumption is obtained in an ASA B 38.2 performance test. The refrigerator is then operated in the humidity room selected for a period varying from one month to twelve months and the B 38.2 energy consumption test repeated. Correctly designed and constructed refrigerators should obviously show no increase in power consumption due to an increase in cabinet heat leakage resulting from moisture entrance into the insulation space.

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31. SMALL REFRIGERATING UNITS

THE term mechanical condensing unit is applied to a complete high side of a small refrigeration system, from ⅛ hp up. It is produced by a score of companies and evolves from both industrial and household machine practice. The refrigerants used are sulfur dioxide, methyl chloride, Freon-22, Freon-12. The single-acting reciprocating type compressor is used almost entirely, for both domestic and commercial applications, although the rotary unit also finds use (**Fig. 1**). Mounting a compressor with its drive and motor, condenser-receiver and controls on a common base has been common up to 75 or 100 hp.

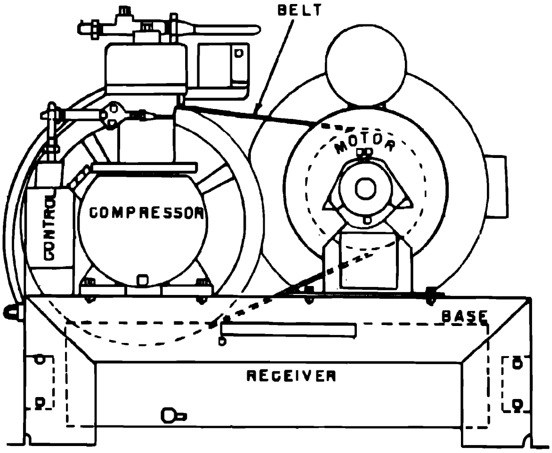


Fig. 1. Typical Commercial Condensing Unit

Air-cooled condensers usually are confined to units below 3 hp. In small sizes, two-cylinder compressors predominate. Larger units tend to favor either three cylinders in line, a four-cylinder V-type or three, six, and eight cylinder W-type arrangements. There are compressors with four cylinders in line, but it has been found that such units tend to develop more vibration than the V-type. V-belt drives are common, which may or may not be provided with automatic adjustable motor bases for proper belt tension. In the smaller sizes, it has been found satisfactory to use a multiple-belt drive which gives a very low loading to the belt, and any stretch is kept to a minimum. With such drives there is no need to apply an automatic belt take-up. Compressor speeds range from 300 to 700 rpm, but higher speeds up to 1,750 rpm are used. Motors are of high torque, low starting current and in most instances run at 1,750 rpm. Compressors fitted with unloaders can start with normal torque motors.

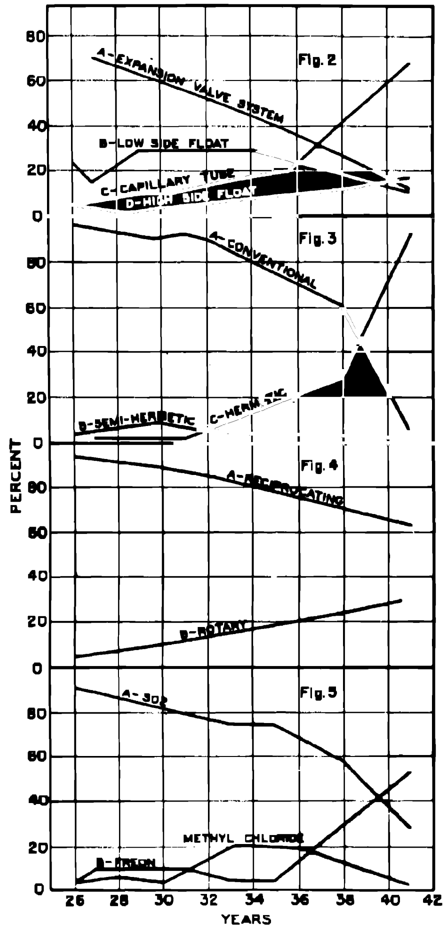
Domestic Refrigeration Units

1. Small automatic self-contained refrigerating systems as used for domestic purposes can be classified according to the type of control: (a) high-side float; (b) low-side float; (c) expansion valve; and (d) capillary tube (see Chap. 22).

Fig. 2 shows the approximate extent of use of each type, by years. Another classification is those using reciprocating compressors, against those using rotary compressors.

Still another classification is made with regard to sealing—between the conventional system, the semi-hermetic system, and the hermetically sealed system.

The **semi-hermetic system** is a compromise between the conventional and the hermetic systems. The main difference in this unit from the conventional type, is that it operates at the full speed of the motor and both motor and compressor are contained in a gas-tight housing. Provision is made in the semi-hermetic unit for access to valves and pistons so that some repairs can be made in the field. However, the fits and tolerances in both the semi-hermetic and hermetic are such as frequently to make it impossible for the usual field shop to do a satisfactory job. Units of this type should be returned to the factory for repair.



Figs. 2, 3, 4, 5 (top to bottom). Extent of Use of Various Types of Domestic Refrigeration Units

Fig. 2. Classified according to type of control
Fig. 3. Classified according to sealing
Fig. 4. Reciprocating or rotary compressors
Fig. 5. Classified according to refrigerant

The **hermetic system** has its motor and compressor completely contained within a sealed housing. No repairs can be made in the field, as it is necessary to cut this housing in order to gain access to the interior. Systems using hermetic and semi-hermetic units for domestic refrigerators are fabricated so that the complete high-side and low-side equipment is connected, sealed, dehydrated, charged and tested as a finished assembly before mounting within the refrigerator compartment.

2. Trends in compressor design. The **reciprocating compressor** will give the best overall efficiencies because of the more positive action of its component parts. It has the best volumetric efficiency, the lowest leakage loss per unit of volume handled, and the best power consumption of any other compression means. The reciprocating compressor, however, is at a disadvantage when extremely large displacements are required, because cylinder bores and strokes or speeds then have to be excessively large.

The reciprocating compressor has less leakage because of the increased effective sealing surface between the piston and cylinder, aided by oil as the sealing agent. It also lends itself to manufacturing because the design is adaptable to the usual drilling, boring, honing, milling and broaching operations.

The **rotary compressor** has worked out very satisfactorily for the small domestic assembly because of its fewer running parts and its adaptation to high speed production. It is not as efficient as a reciprocating unit, but if exceptionally large condenser area is used, lower head pressures result and the design is very satisfactory. The rotary compressor takes very little space, can be operated at high speeds and starts very easily.

The conventional reciprocating compressor, driven by a belt at moderate speeds, is usually lubricated by splash lubrication. Semi- or full-hermetic compressors, either rotary or reciprocating, are usually lubricated with a combination of splash and force feed. Some designs use a modified form of gear pump which conveys the oil to the running parts, while other designs use a form of rotary pump on the end of the shaft, dipping into the oil sump and flooding working parts with a bath of oil.

The rotary compressor does not use valves but the reciprocating compressor uses a reed-type valve assembled in a valve plate in the cylinder head. Some types of reciprocating compressors vary from the conventional design; that is, the cylinder is actuated by the crankshaft while the piston is held stationary. Some types combine the connecting rod and piston by using an oscillating cylinder, and others use the Scotch yoke drive rather than the conventional crank or eccentric.

Two types of rotary compressors are being used at the present time. One type incorporates a rotating eccentric and one blade which is held in the compressor housing. The other type uses a rotating shaft which holds two or four blades, properly spaced, and rotates them in an eccentric housing. The oil performs an important function in maintaining a good seal in all rotary compressors.

Refrigerants

(See also Chap. 8)

3. **Sulfur dioxide** was once the most popular refrigerant (**Fig. 5**), although giving way to others in recent years. Factors favoring sulfur dioxide are that it is very stable, it simplifies lubrication problems, has good specific volume, effective latent heat, and good operating pressure characteristics. However, due to its irritating odor and corrosiveness when combined with water, its greatest use is in hermetic units.

Methyl chloride operates at higher pressures and has considerably less corrosive action in the presence of moisture, a good operating range, and does not require special compressor designs.

Freon-12 is popular because of its low specific volume which permits a small displacement compressor. When evaporator temperatures are very low (approach −40 F), it is desirable to use **Freon-22,** which has characteristics resembling those of ammonia and which permits the low-pressure designed compressor to develop lower evaporator temperatures.

Freon-22 has recently been taking the place of **Freon-12** in many general applications such as air conditioning, water cooling, etc. The reason being that about $33\frac{1}{3}\%$ less displacement for the same capacity can be secured. Thus in many instances a lower cost compressor is possible than would be necessary with **Freon-12. F-21, F-114** and **methylene chloride** are sometimes used in rotary units. **Isobutane, ethyl chloride** and **methyl formate** refrigerants have been used occasionally. **Ammonia** is used in the absorption machine.

The **lubricant** used in domestic refrigerating machines has been the subject of considerable study. The oils now used are usually of the mineral base and must be chemically pure, with special attention paid the allowable maximum amounts of water and sulfur contents. The viscosity requirements are similar to those for small commercial machines, differing somewhat due to the refrigerants used because of the thinning action produced in the crankcase by dilution with the refrigerant. Oil must be heavy enough to provide a good body at higher temperatures and still remain fluid at temperatures at least 10 to 15 deg lower than the lowest temperature likely to occur in the evaporator. Shown below are desirable oil **viscosities** for the three most popular refrigerants, S. U. at 100 F.

Sulfur dioxide	75–100
Freon-12	150–200
Methyl chloride	150–300

4. **Controls.** The pressure-operated **expansion valve** is popular for domestic refrigeration as it takes up very little room and is easily adjusted. However, it is not easy to make such a valve cheaply without compromising its performance, and for this reason it is not used as much as formerly. This type of valve also presents a hazard in connection with a hermetic system, because the system must be opened to the atmosphere to repair such a valve.

The **low-pressure float** has the advantage of simplicity of construction, operation and manufacture. Also, the float may be part of the evaporator and it is adapted to quantity production.

The **high-pressure float** in its early design was too cumbersome for domestic refrigeration application, but with the growth of the package hermetic system and refinements in design, it has become increasingly popular. To mount the high-side float remote from the cooling coil necessitates the use of an auxiliary refrigerant control to prevent the liquid line from frosting between the control and the cooling coil. Two of these auxiliary controls are in common use, namely, a **capillary tube** or weighted **check valve.** These secondary throttling devices permit the installation of the liquid line at room temperatures without the possibility of frosting or sweating. With the high-pressure float system, very accurate control of the refrigerant charge must be maintained; otherwise a very slight overcharging will cause excessive head pressures.

The **capillary tube** control system has many favorable features, particularly with hermetically sealed domestic units, because of simplicity of construction, low cost, absence of moving parts and the permitting of balanced pressures for ease in starting. When capillary tubes are used for refrigerant control, the general practice is to use a suction accumulator which prevents liquid spill-over to the compressor and thus eliminates difficulty in starting of the condensing unit. No receiver is used with a capillary system. **Table 1** shows the general practice of capillary tube size for various capacities.

Table 1. Capillary Tube Data for Various Conditions (Freon-12)

hp	Btu/hr	Suction temp F	Size of accumulator, cu in.	Capillary tube	
				Bore, in.	Length, ft
$\frac{1}{8}$	800	18	8	.032	6
$\frac{1}{6}$	975	18	8	.040	17
$\frac{1}{5}$	1,550	26	8	.040	6
$\frac{1}{5}$	1,550	26	8	.050 .040	9½ 4
$\frac{1}{4}$	2,125	26	12	.050	6½
$\frac{1}{4}$	2,125	26	12	.040	3
$\frac{1}{3}$	2,800	26	12	.050	5
$\frac{1}{6}$	800	0	8	.040	15
$\frac{1}{4}$	1,250	0	12	.040	12
$\frac{1}{3}$	1,620	0	12	.040	10
$\frac{1}{3}$	1,620	0	12	.050	12

5. Where quietness is essential, as in the domestic unit, the **capacitor-type motor** has proved most acceptable. Its characteristics lend it to application in hermetic-type compressors. The cooling of the motor is usually by fins on the housing cover, by circulating oil over the motor windings, or by permitting the cool return suction gas to come in contact with the windings. Considerable research was expended to discover a satisfactory insulation for the motor windings exposed to the refrigerant. Neoprene-coated wire, plain copper wire insulated with bleached long-fiber cotton, and a synthetic enamel known as Formex, have proved most successful. The assembly must be very thoroughly dehydrated and baked under a vacuum.

All domestic machines use a **motor control** to permit the automatic regulation of temperature. Two types have been used, the pressure-actuated and the thermostatic, the latter being more popular because of more accurate regulation. The newer controls, in addition to their motor control function, usually incorporate a manual switch, a calibrated temperature adjustment, an overload cutout and a defrosting switch.

6. The early domestic **evaporators** were made of copper tubes spun around a tray compartment, for direct expansion or flooded control. The trend was later toward pressed steel parts, hydrogen brazed. This design made it quite simple to fabricate serpentine cups in steel, so that when the parts were pressed together, they formed passages for the refrigerant. While this construction permitted fabrication in punch presses, there was difficulty with rusting until porcelain coatings and stainless steel materials were specified.

In design, the humidities and temperatures necessary to handle stored foods must be kept in mind. Generally, the temperature maintained in the evaporator is 18 F and in the refrigerator from 38 to 45 F, with relative humidity of from 70 to 80%. Since so much is expected of a domestic refrigerator, from making ice cubes and ice cream to preserving various foods, some dehydration is unavoidable. Special compartments to maintain proper temperatures and humidities are used.

All refrigerators are thermostatically controlled, either by having the temperature bulb of the thermostat strapped to the evaporator or to one or the other of the compartments in the box. The **thermostat** is provided with means for regulation of box temperature above or below the normal setting. Arrangements are made to defrost the evaporator periodically by shutting off the compressor. Some controls automatically restart the unit after defrosting for about an hour or two.

Many of the newer refrigerators use hot gas defrosting with timers set to operate every evening. Thus the boxes and coils are always at peak efficiencies.

7. The newest model refrigerators have space where **frozen foods** may be stored at temperature around 0 F. This imposes still further temperature regulations upon the condensing unit. If a single condensing unit is to maintain both frozen food and medium temperatures, obviously additional controls will have to be provided to regulate these diverse requirements. The condensing unit will have to operate at the lowest temperature and be fed and maintained at suitable medium temperatures for other services in order not to reduce humidity.

Some dual-service refrigerators utilize a secondary refrigerant circuit to develop medium temperatures which is claimed to simplify the controls and give better regulation. Many of the new refrigerators have coils surrounding the inner liner so the usual evaporator does not appear. The liner type of evaporator tends to improve relative humidity because of less air movement and reduced temperature differential between air and refrigerant.

Commercial Compressor Performance

8. The theoretical **performance** of a compressor in a condensing unit is the same as that of any compressor, as discussed in Chaps. 2 and 23. It is possible to predict performance of compressors for any gas. Understanding of compressor performance can be had by reference to the $P-v$ relationship of the indicator card. The theoretical **volumetric efficiency** of a compressor is discussed in Section 2 of Chap. 2.

It is obvious that if n is small, volumetric efficiency will be lower for the same per cent clearance volume. The value of the exponent n for several refrigerant gases is given in **Table 2.** It is seen that low-pressure refrigerants have rather low values of n. It can be expected that the volumetric efficiencies will be of a lower order than air or ammonia. Careful attention must be given to the reduction of excess cylinder clearances.

The theoretical **power** required to compress a cubic foot of gas is expressed as follows:

Table 2. Value of Specific Heat Ratio

($C_p/C_v = n$) for some refrigerant gases

Air	**1.40**
Ammonia	**1.31**
Carbon dioxide	**1.3**
Ethyl chloride	**1.19**
Freon-12	**1.13**
Methyl chloride	**1.20**
-ButaNne	**1.11**
Propane	**1.13**
Sulfur dioxide	**1.29**
Freon-22	**1.20**

Theoretical adiabatic horsepower

$$= \frac{144}{33{,}000} \frac{n}{n-1} P_1 v_1 \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] \quad (1)$$

Where,

P_1 = psia suction pressure

v_1 = cu ft of gas entering compressor section

If the formula is based on one cubic foot, v_1 is eliminated, and the theoretical adiabatic horsepower becomes

$$\text{thp} = P_1 \left\{ \frac{144}{33{,}000} \frac{n}{n-1} [(R)^{(n-1)/n} - 1 \right\} \quad (2)$$

$\text{thp} = P_1$ (hp constant).

Values of horsepower constant are plotted in **Fig. 6,** for various compression and specific heat ratios. To obtain the adiabatic horsepower per cu ft, it is necessary to multiply the horsepower constant by the absolute suction pressure.

9. Experience indicates that the theoretical volumetric efficiency will give only a rough approximation of the true value because of many factors which influence it and which are difficult to measure or predict. Further tests indicate that leakage plays such an important part and is so difficult to measure that any extensive practical analysis must rest on actual test. The effect of cylinder superheating has been discussed by Crampton.[1] In design, the probable volumetric efficiency can be calculated. The actual volumetric efficiency will be as much as 15% less. On open-type compressors, the displacement can be estimated and arrangements should be made to adjust the overall capacity by increasing speed as developed by actual tests. (See Chap. 2 and 23.)

A composite curve of volumetric efficien-

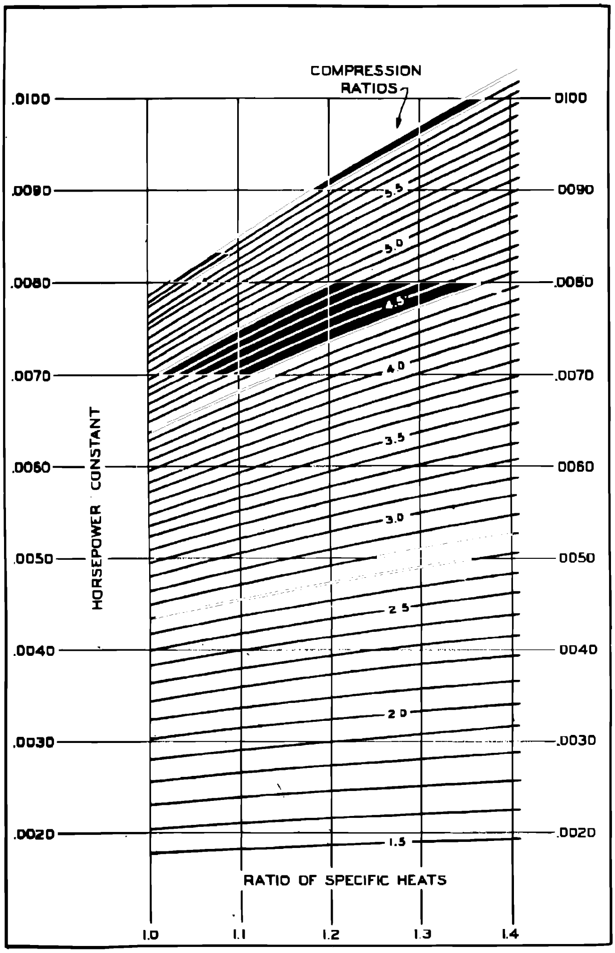


Fig. 6. Horsepower Constant for Various Compression and Specific Heat Ratios

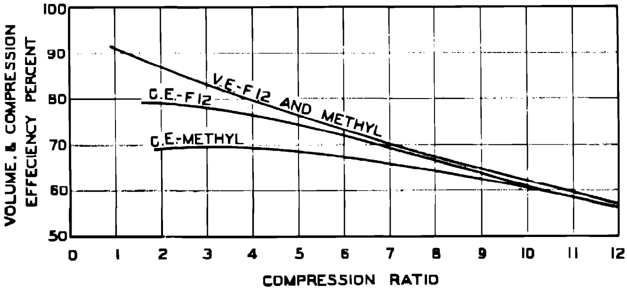


Fig. 7. Volumetric (V.E.) and Compression (C.E.) Efficiencies of Compressors

cies of all compressors would be hard to define, but a representative curve which gives very close results is shown in **Fig. 7**, for Freon-12 and methyl chloride. Values for sulfur dioxide would approximate the methyl chloride curve. An important factor in testing for volumetric efficiency is superheating of the suction gas, which will be considered in more detail later.

The power required to drive the compressor is found by the following:

$$\text{bhp} = \frac{\text{thp} \times d \times e_1}{e_2 \times e_3} \tag{3}$$

Where

bhp = brake horsepower

thp = theoretical adiabatic horsepower per cu ft

d = displacement of compressor, cfm

e_1 = volumetric efficiency, $\frac{\text{percent}}{100}$

e_2 = compression efficiency, $\frac{\text{percent}}{100}$

e_3 = mechanical efficiency, $\frac{\text{percent}}{100}$

Representative values of compression efficiency are shown in Fig. 7, for Freon-12 and methyl chloride. Mechanical efficiency should not be constant, but should depend upon the actual loading of the compressor at the conditions. It would be preferable to find the friction horsepower first, or estimate it. If the compressor were designed for a 5-hp maximum load, the friction horsepower could be expected to be about 0.40, based on a peak load mechanical efficiency of 92%. If the compressor were only loaded to 3 hp, at the same speed, the friction horsepower should be 0.40 and the mechanical efficiency would be only 88%. The friction horsepower will vary almost directly with rpm.

Since practically all condensing units must start against pressure, it is desirable to estimate the starting torque required in lb ft. Since the following equation is for estimating purposes, it neglects the friction of the compressor, the additional torque required to force discharge gas out of the cylinder, and the fact that the tangential force at the crankpin is not always equal to the normal force at the piston.

$$T_s = \frac{(P_2 - P_1)As}{24 d_1/d_2} \tag{4}$$

or

$$T_s = \frac{(P_2 - P_1)As}{24\, r_2/r_1} \tag{5}$$

Where

T_s = starting torque, lb ft
P_2 = discharge pressure, psia
P_1 = suction pressure, psia
A = area of cylinder, sq in. (Use only one area for a two- or four-cylinder compressor.)
s = stroke of compressor, in.
d_1 = diameter of compressor pulley, in.
d_2 = diameter of motor pulley, in.
r_2 = motor rpm
r_1 = compressor rpm

This formula shows that the direct-connected compressor will need more motor torque to start it than a belted arrangement. The starting torque available in a motor is obtained as follows:

$$T_s = \frac{5{,}250 \text{ hp}}{\text{rpm}} \; \frac{B}{100} \tag{6}$$

Where

T_s = starting torque, lb ft
hp = horsepower rating of motor
rpm = full load speed of motor
B = percentage starting torque = 400 to 600% for single-phase motor, or 225 to 325% for polyphase.

Actually, the starting torque is too all-inclusive a term and should be properly broken down into its two major factors, breakaway torque, or the torque to start the compressor from a dead stop, which is more nearly that given by the formula, and a pull-in torque which is the final torque of the motor on its starting windings just prior to its transfer to full load running conditions. The pull-in torque of the motor must always be sufficient to accelerate the compressor from rest, overcoming both inertia and gas forces to bring itself to synchronous speed. It is probable that the pull-in torque of a loaded compressor may require double that at break away. In medium size compressors, because of heavy current draw, the practice of unloading at start by holding open suction valves has become more prevalent. By unloading only 50%, it is possible to allow use of normal starting torque motors.

Because of the difficulty of obtaining indicator cards for small compressors, breakdown efficiencies are seldom attempted. Actual tests are essential and the comparisons are usually made on the basis of capacities in terms of Btu per hr capacity per watt hr of motor input. Representative curves of the performance of a ½-hp air-cooled condensing unit are shown in **Fig. 8.** Typical manufacturers' performance specifications are shown in **Table 3.** Since, at low suction pressures, the compressor needs more displacement to load the motor, it is commercial practice to split the operating temperature range into three sections, usually designated as low, medium or high temperature, and operate the compressor at three different speeds, corresponding to each range. Sometimes, a larger compressor is necessary to satisfy the low-temperature ranges. This makes possible the most effective use of the available motor power.

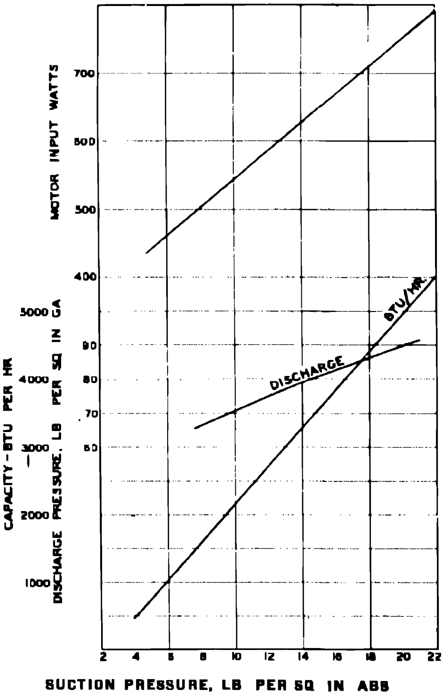


Fig. 8. Air-Cooled Sulfur Dioxide ½-hp Compressor, 2-cylinder, 2¼ in. bore × 1⅝ in. stroke, 510 rpm, 90 F room

The hermetic compressor, because of inability to adjust its speeds to such operating ranges, cannot produce as much capacity at low temperatures as the belt-driven unit. This is because the hermetic unit is normally designed for high suction pressures and motors are applied for that rating. This influences design in many ways, as explained previously; that is, when the capillary tube is used, balanced pressures at the start result in reduced starting requirements.

Table 3. Typical Specifications, 1-hp Air-Cooled Condensing Unit

(2-cyl, 2½-in. bore × 1¼-in. stroke, 1-hp motor, 2 V-belts; overall dimensions 32 × 24 × 21 in. high.)

Condenser	Continuous fin
Receiver capacity, lb, methyl chloride	17
Receiver capacity, lb, Freon-12	25
Liquid line fitting	⅜ in. S.A.E.
Suction line fitting	⅝ in. S.A.E.
Oil charge, pints	3¾
Weight, lb	294

Capacity with methyl chloride, 90 F ambient

Evap temp, F	rpm	Capacity Btu/hr
−40	640	1,900
−30	640	2,700
−20	640	4,050
−10	640	5,500
0	640	7,390
−10	530	4,650
0	530	6,050
10	530	7,800
20	530	9,750

Capacity with Freon-12

Evap temp, F	rpm	Capacity Btu/hr
−40	570	2,775
−30	570	3,775
−20	570	4,900
−10	570	6,350
0	570	8,100
−10	500	5,200
—	500	6,700
10	500	8,300
20	500	10,200
30	365	9,225
40	365	11,050
50	365	13,100

Compressor Design

10. Low-pressure refrigerants permit less restrictions regarding **materials** than ammonia. All forms of steel, brass, copper, zinc, aluminum and die castings can be used with Freon-12 or sulfur dioxide. The same is true of methyl chloride with the exception of aluminum. There is a very pronounced chemical reaction between aluminum and methyl chloride and, therefore, these two materials should never be used together. Die castings in which a certain percentage of aluminum is used have been a part of methyl chloride systems with apparently no ill effects.[2,3]

To design a compressor for low-pressure refrigerants requires a knowledge not only of the gas handled and its characteristics, but the way in which the actual system functions. It is not only necessary to provide the proper cylinder proportions and valve openings, but also the proper arrangement for returning gas and oil to the cylinder.

11. A sulfur dioxide compressor will have certain differences from a Freon-12 or methyl chloride compressor, while the differences between Freon-12 and methyl chloride will not be apparent. These differences center about the pressures involved and the affinity of the gas for the oil used. Sulfur dioxide operates at below atmospheric suction pressures in most instances and is very critical regarding oil pumping. It is desirable to have some form of catch chamber to separate the oil slugs and return them to the crankcase, before the gas is admitted to the cylinder. It is usually desirable to operate the sulfur dioxide compressor at a somewhat slower speed, in order to give added protection against damage to the compressor. While it is true that the hermetic compressor operates on sulfur dioxide at high speeds, this application is mostly restricted to the domestic unit, where small capacities can be more readily handled. In addition, large separating chambers are provided to give the necessary protection.

Freon-12 and methyl chloride compressors must also be provided with arrangements to permit all oil that returns with the suction gas to be drained to the crankcase. Experience indicates that the requirement for separator chambers on the suction is not so critical as with sulfur dioxide. This is apparently due to the complete mixing of the oil with the gas, permitting a more uniform and constant return at all times.

12. The design of **crankshaft** and **bearings** follows the usual compressor practice. It is usual to consider 200 psi pressure difference as the basis for Freon-12 and methyl chloride piston loads. If condensing units are to pass the Underwriters Laboratories' tests, the crankcases must be designed for hydraulic test pressures of five times the refrigerant pressure at 70 F, and the cylinders, heads, and all high-side fittings at five times the setting of any pressure limiting device. The usual practice is to test both low and high sides at least to 195 psi pressure before shipment.

13. It is common practice in many designs to harden the **shafts** to a Scale C Rockwell index of about 50 to 60. The hardened shaft serves as a riding surface for the seal and provides a superior wearing surface for bearings, especially where leaded bronze bearings are used. Experience indicates the inadvisability of using a crankshaft having any chromium content

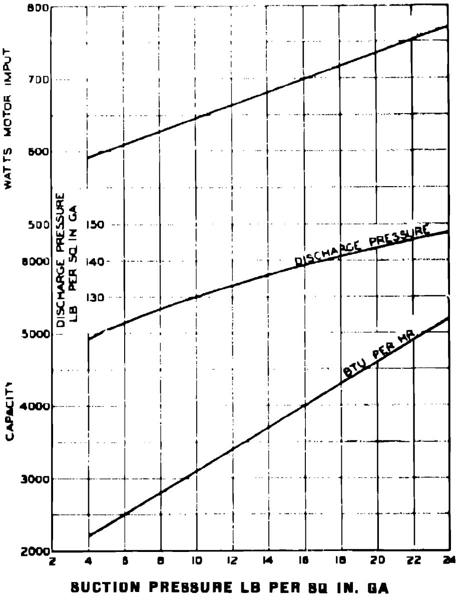


Fig. 8a. Air Cooled Freon-12, ⅓-hp Compressor, 2-cylinder, 1-13/16 in. bore×1⅜ in. stroke, 525 rpm, 90 F room

if a shaft shoulder is used as a sealing surface. Chromium specks in the steel cause an imperfect surface for the seal. Steel of Class SAE X3140 should be used only where seals do not ride directly on the shaft. SAE X1315 steels properly case-hardened are better for shaft riding seal designs.

14. Diaphragm and bellows **seals** have been most popular until recent years. A construction which eliminates the bellows and instead floats the seal nose in a synthetic rubber shaft sealing collar, while the nose bears against a stationary seal plate, has proved very satisfactory and is gaining wider acceptance. Seal noses have been made of special bronzes and metal impregnated carbon. In seal designs, when the shaft diameter exceeds 2 in., it is advisable to use a multiple-spring design to maintain contact of seal against the shaft. This is especially true of the type of seal which comprises a separate hardened or alloy cast iron nose-piece, sealed to the shaft with a Neoprene or synthetic rubber gasket. This type tends to tip or cock, unless the nose piece is compressed evenly.

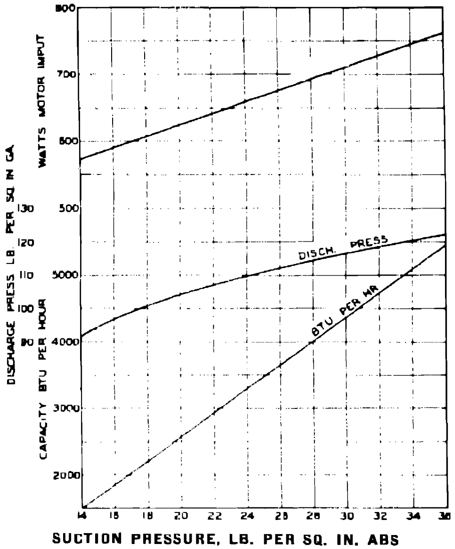


Fig. 8b. Air Cooled Methyl Chloride $\frac{1}{2}$-hp Compressor, 2-cylinder, 1-13/16 in. bore × $1\frac{3}{8}$ in. stroke, 545 rpm, 90 F room

Engineers are varied in their opinions regarding a one-piece crankcase and cylinder. Properly tooled, a good shop can handle either effectively. There is no difficulty in securing alignment with the divided construction. The cast iron used must be dense and close-grained, preferably with a nickel content of from 1 to 2%.

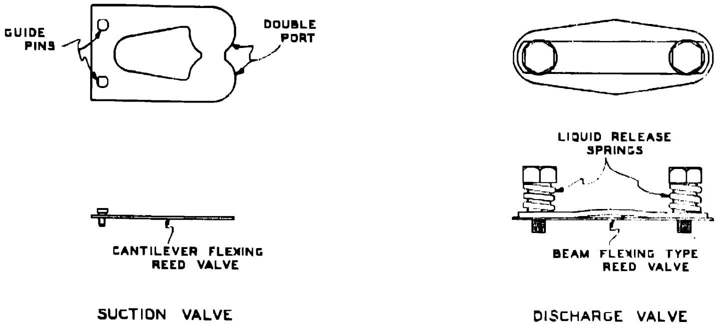
It is not advisable to provide eccentric drives for strokes greater than 4 in. Static friction of a larger diameter eccentric increases the starting torque requirements.

15. Cast iron pistons are used in most designs since they permit extremely close fits and give good efficiencies. In small compressors, leakage may become a large percentage of the total gas pumped and close tolerances must be observed, with all pistons and cylinders ground and honed to mirror finishes. The usual piston and cylinder clearance is 0.0003 in. per in. of cylinder bore. There is no advantage in using rings on pistons smaller than 2-in. bore. The plug type, no-ring piston is common practice in domestic and commercial units of $\frac{1}{2}$ hp and smaller.

Piston rings, where used, are generally fitted between top of piston and wrist pin, according to automotive practice. Two compression rings and one oil ring are used, although in Freon-12 and methyl chloride units an oil ring is hardly needed, as oil flow through the system is encouraged. The trunk-type piston design admits suction gas through the center of the piston, and may have an additional compression or combination compression and oil wiper ring on the piston skirt. The trunk-type piston design is especially useful in high-speed compressors, as placing the valve in the piston gives greater space for compressor valves, permitting lower valve velocities. Where moderate speeds are used, suction and discharge valves can be located in a common head plate and this construction provides certain desirable simplifications in servicing.

16. For small compressors, high volumetric efficiencies can only be obtained by small re-expansion volume. **Valve velocities** for both suction and discharge valve at about 3,000 fpm should not be exceeded, especially for Freon-12 units. For estimating purposes the velocity calculation is

$$V=\frac{2\,S\times \text{rpm}\times \text{piston area}}{12\times \text{area of valve opening}} \qquad (7)$$



SUCTION VALVE

DISCHARGE VALVE

Fig. 9. Reed Valves

where S = stroke, in.

The actual valve velocity must be based on the actual percentage of the stroke during which time the valve is open. Ingenious designs are required to get these velocities at minimum re-expansion volume. The reed valve, **Fig. 9**, works well on small units.

In larger compressors, the ring-plate or disc valve gives performance and durability which cannot be obtained with the reed type, where the suspension between supports becomes too great (**Fig. 10**). Swedish steel gives best durability and grain structure for valve elements. Utilizing the usual bending formula, for disc and flat plates and for flat springs as beams, it is advisable to limit the allowable working stress to 30,000 psi. A good practice to follow in all valve design is to prevent the reed, disc, or plate from impinging on any backing springs, either flat or coiled, which form a part of the valve assembly. A stop to prevent a curved spring from being compressed flat, or a coil spring being compressed until it meets an adjacent coil, should be provided.

Some designers provide a liquid relief in the discharge valve, opening when liquid slugs enter the cylinder.

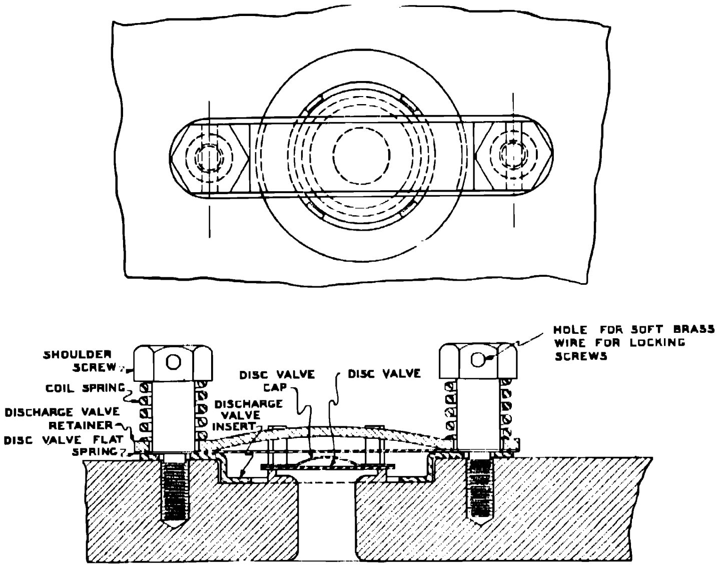


Fig. 10. Discharge Valve Assembly

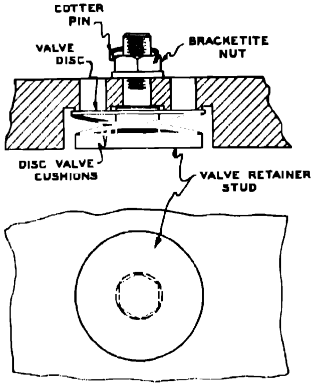


Fig. 10a. Suction Valve Assembly

It has often been the practice to consider area through the seat all important, while lift area is ignored. This is a great mistake. It is desirable to have lift area approach seat area as closely as possible in order to obtain maximum volumetric and compression efficiencies. The lift area should be at least 80 to 90% of the seat area. This cannot be done with a single valve in most cases without having a high lift which will cause heavy impacts and short life for the parts. By breaking the valve requirements up into a number of small valves however, lift will be reduced.

For example, for a circular disc valve:

$$\text{Seat area} = \frac{\pi D^2}{4} \qquad \text{Lift area} = \pi DH$$

$$\text{and } \frac{\pi D^2}{4} = \pi DH \quad \text{Therefore } H \text{ (lift)} = \frac{D}{4}$$

If we use two valve discs of a new diameter d, then

$$\frac{\pi D^2}{4} = \frac{2\pi d^2}{4} \qquad \text{and} \qquad D = d\sqrt{2}$$

The new lift, h, will give a lift area $2\pi\ dh$ which will equal $\pi D H$.

$$2\pi dh = \pi DH$$

$$2dh = DH$$

but

$$D = d\sqrt{2}$$

$$2dh = dH\sqrt{2}$$

$$2h = H\sqrt{2}$$

$$h = \frac{H}{2}\sqrt{2}$$

or by using two discs to split the valve area instead of one, we reduce the height of lift by $\frac{1}{2}\sqrt{2}$.

It is not generally realized that valves can cause considerable noise in a compressor over and above the normal "clicks" which are expected. Sluggish restricted valves can cause rough running gear plus gas noises which are traced with difficulty. Just what occurs to develop this is difficult to state. There may be basis for a comment that a valve that does not close promptly causes severe stress reversals at dead end stroke which rattles the shafts and rods.

Higher and higher speeds necessitate lighter valves to reduce inertia effects, all of which encourages the trend to smaller valve elements. The use of a number of small valves for high speed thus results in better opportunities to obtain large lift and seat areas and lower inertia losses. The valve design alone is the factor which performancewise makes possible equivalent and sometimes better efficiencies than those of the slow speed machine. It is certainly true that neither volumetric nor compression efficiencies can be predicted for a compressor on the basis of the percent clearance or the compression ratios without analysis of the type of valve, the gas velocities through both seat and lift and those surplus expenditures of work needed to compress gas through the required pressure ranges.

17. Streamlining passages or ports, or placing fins on the heads of the cylinders, seems to make negligible changes in efficiencies in sizes below 1 hp; in fact, the effects are only slightly evident in units as large as 50 hp. Even water-jacketed heads are being discarded. Gas passages through the heads should have about the same or lower velocities as those through the compressor valves. In some instances, assumptions have been made that discharge valve and port velocities can be higher than those of the suction valve, because the discharge period is shorter than that of the suction period. When consideration is given to greater friction because of the higher pressure, it is desirable to maintain discharge velocities as low as those of the suction.

Compressors larger than 2 hp require ample gas space above the discharge valves, to iron out violent pulsations. Unless this is done. valves suffer reverse flexes, causing early failure, and compressor efficiencies will be affected. In shell condensers, pulsations are absorbed within the large volume, but with evaporative condensers and long discharge lines, an external expansion chamber must be located at the compressor outlet. An oil separator often serves this purpose.

It has in general been a recommendation that gas pulsations and noise can be ironed out by providing low velocity in the discharge line from a compressor. This often tends to increase the size of line but results are not entirely satisfactory.

A newer theory, giving excellent results, has been to greatly increase the discharge line velocity to considerably higher values but still not sufficiently to materially affect performance. In combination with a discharge muffler, a smaller discharge line can make a great reduction in overall compressor gas noises.

18. **Main bearings** can be of any well-known material, such as cast iron, leaded bronze or steel shell Babbitt. The favored bearing is leaded bronze, as it is non-scoring. Bearing clearances vary between 1 and 2 thousandths per inch of shaft diameter depending on loading. Needle, roller, or ball bearings have been used very successfully in main bearings. Needle bearings are often selected for wrist pin bearings. Such types, however, are somewhat noisy, and because of the usual requirement for quietness, have not been widely applied to condensing units in sizes below 15 hp.

19. **Connecting rods** are of the conventional type and have been specified either as forged or cast iron construction. The cast iron should have certain malleable qualities with sufficient ductility to absorb shock. Eccentric straps, where used, are generally cast completely of a bearing bronze, making it unnecessary to add bushings at the bearings. Wrist pins are hardened and ground, usually of a case-hardened material, such as SAE X1315 and then hardened to a Rockwell Scale C—50–60.

Recent research and experience indicate that to secure least wear between rubbing or sliding surfaces, there should be a Rockwell hardness difference between the two parts of at least 50 points.

20. **Lubrication** of small compressors up to about 25 hp is handled very satisfactorily by splash lubrication, although some manufacturers use force feed even in very small sizes. There is no indication that any reduction in oil pumping or friction horsepower occurs where force feed is used. If speeds are increased beyond 800 to 900 rpm, force feed is desirable because it eliminates violent splashing. Oil pumps must be carefully fitted, in order not to lose prime if gas enters the pump. Gear pumps are favored. In small sizes up to 10 gpm they can be connected to the end of the shaft, as ample lift is developed by a good gear pump to pick up the suction between the base of the crankcase and the center line of the shaft. A strainer should be inserted in the suction line to keep out foreign substances from the pump and bearings.

In designing a cylinder, some provision must be made to return oil from the suction port to the crankcase. As there is always leakage of gas upward from the crankcase to the port, the velocity must be low in order that oil may drop against the upward gas flow. At least 5% of the total piston displacement is leakage. This must escape through the return oil opening at a velocity not to exceed 200 fpm. A separating chamber can be built into the cylinder to help separate oil from the gas, before it enters the suction port.

In many designs, a check valve is inserted at the bottom of the oil return port to prevent a surge of crankcase oil entering the suction. A bypass must be provided in this check valve, which is always open, to permit the check valve to open wide after the oil surge has passed. Otherwise, the check valve would close tightly. When a separating chamber is used, the oil surge is trapped before it can enter the suction port, and a check valve is thus not essential.

21. **Regulation** of compressor capacity can be accomplished in three ways:

1. By throttling suction
2. By bypassing cylinders
    a. Using an external solenoid bypass valve
    b. By holding open the suction valves
3. By variable speed motors.

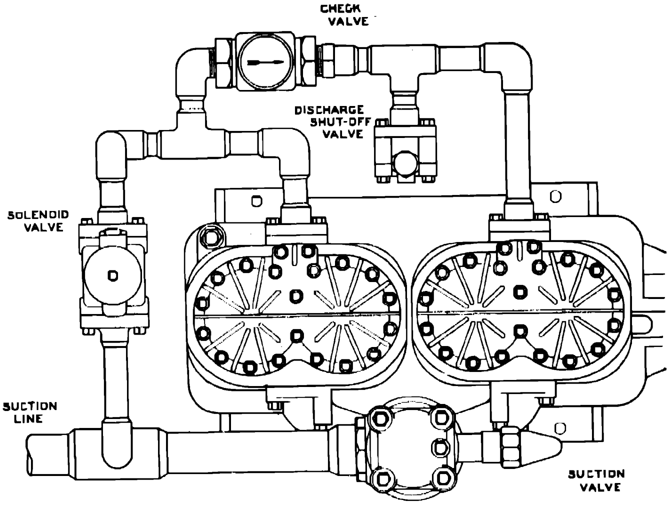


Fig. 11. Cylinder Cutout Control Piping

Throttling the suction is inadvisable, because pulling down the suction pressure causes crankcase oil to froth and be pumped from the compressor. If violent enough, it can break valves or bend connecting rods. In addition, operation at reduced suction pressure is uneconomical in power consumption. Therefore, the first method is seldom used, but in combination with the second gives a rather close regulation, reducing the hazard of throttling.

The **solenoid** is often connected as shown in **Fig. 11.** Care must be exercised to mix the hot bypass with cool return gas, else excessive cylinder temperatures will be built up in the inoperative cylinder. This method is used to give 50% capacity reduction. The check valve should have a discharge area equal to that of the compressor discharge valve. It is a good plan to use the same type of valve in the check as that in the compressor, it being less noisy than a swing check.

Power requirements at 50% capacity will run from 65 to 90% of the full load requirements, depending on the free passage provided through the bypass. Method 2b eliminates the need for external bypass piping and a check valve, as shown in **Fig. 12.** There are other complexities, however. A conventionally designed suction valve must be provided to permit attachment of unloading fingers. These fingers are actuated by a plunger, and the pressure to this plunger is controlled by a three-way solenoid valve, actuated by a temperature or pressure switch. The cylinder operates cooler than with method 2a and can be applied on a four-cylinder compressor to 75% unloading with only one cylinder operating. There must be at least one cylinder

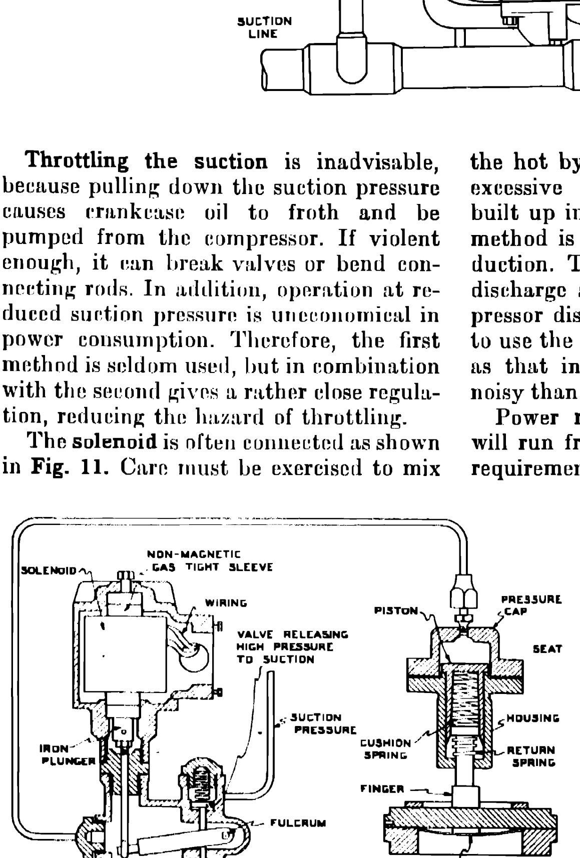


Fig. 12. Use of Solenoid Valve without Bypass Piping

in operation all the time, in order to provide condenser pressure to actuate the unloader plungers. One danger is that the unloader fingers snap down on the suction valve and add stresses to an already overworked part. The force of the blow has also to be absorbed by whatever backing retainer holds the suction valve in position. Power requirements will be about the same as those for method 2a.

The **variable speed motor** is the most efficient means of varying capacity. Usually a two-speed motor, giving 50% capacity reduction, is sufficient for most applications. Power requirements for 50% capacity will be from 55 to 60% of the full load.

In all these unloading methods, to insure an adequate return of oil to the compressor at all times there must be no traps in the evaporator or the suction line. Gas velocities in the suction line at reduced capacity must be above 1,000 fpm.

Commercial Motors

22. The majority of **single-phase motors** used on condensing units are of the repulsion-induction type. This motor has low current inrush per horsepower and extremely high starting torque. These factors favor this design although there has been a trend in some instances toward the capacitor-start induction-run motor. Inconclusive service records show that capacitor motors have lower maintenance cost. As for efficiencies, equal performance can be expected. A capacitor motor larger than 1 hp draws an extremely heavy starting current, which may affect contact points of pressure and temperature controls (see Chap. 20).

Polyphase motors are of the double squirrel-cage type, having high starting torque, with low starting current in-rush. Two-speed motors are usually of the constant torque, single-winding or consequent pole type, 1,800 to 900 rpm or 1,200 to 600 rpm. Dc motors should be compound wound. If specified with inter-poles, better commutation is assured.

23. **Controls.** Single-phase motors, up to and including 1 hp, are usually furnished with built-in overload elements. In larger sizes, or on polyphase motors, across-the-line starters are provided with overload and under-voltage protection, by the use of thermal elements. Built-in over-load protection is of the automatic reset type, but starters have to be reset by hand.

It is not desirable to furnish **automatic resets** in a starter without some form of temperature device buried in the windings of the motor, because the overload elements might reset while the motor is still at a dangerous temperature. It is possible, however, to get motors in very large sizes with built-in thermal elements controlling external starters, thereby obtaining the desirable features of automatic reset.

The usual practice in refrigeration is to provide **overload elements** which will trip after a period of two hours, at 140% of the full load current of the motor, when the motor and starter is of 5 hp and below, and operating in an ambient temperature of 40 C (104 F). For motors larger than 5 hp, overloads are selected at 125%.

Automatic operation is usually accomplished by bellows or diaphragm-type **switches.** Water-cooled condensing units are supplied with a dual control, one part of which, the high pressure cutout, will stop the motor if the condenser pressure becomes excessive. This may occur if there is any lack or stoppage of water. The second part of the control, low-pressure or temperature cutout, starts or stops the motor, depending on the condition at the evaporator.

The air-cooled condensing unit under 3 hp is seldom furnished with a high-pressure cutout because air is always present to prevent excessive head pressures. The overload protection provided in the motor is a sufficient safety limiting device; because of fouling of the condenser, there will be excessive current drawn from the line, eventually stopping the motor. With this design, a straight pressure or temperature control regulates the motor operation.

In room temperatures below 20 F, it may be necessary to restrict air flow to the condenser. This should only be done if a dual control is specified, in order to protect against too high head pressure. If the head pressure is too low, insufficient pressure difference is developed to force liquid through the expansion valve, unless the orifice is made excessively large. With low room temperatures a pressure control can-

Table 4. Compressor Ventilation Requirements

Compressor hp	Natural circulation, cu ft room volume needed		Forced circulation, cfm fresh air entering room	
	Air-cooled	Water-cooled	Air-cooled	Water-cooled
¼	200		100	
⅓	400	190	145	130
½	600	260	220	175
¾	800	360	330	275
1	1,000	475	440	300
1½	1,500	665	660	425
2	2,000	850	880	550
3	3,000	1,100	1,320	780
5		1,300		1,100
7½		1,450		1,400
10		1,600		1,700
15		1,850		2,250
20		2,000		2,750
25		2,150		3,150
30		2,250		3,500
40		2,500		4,250
50		2,750		5,000

not operate, since the suction pressure is below that corresponding to room temperature. In such instances, it is desirable to use a temperature control to maintain evaporator temperatures.

Air-cooled units must be furnished with sufficient ventilation to maintain their efficiencies. Table 4 indicates the minimum size of room in which air or water-cooled condensing units should be installed. When smaller rooms are used, some form of forced ventilation should be provided.

24. Tests indicate that dry gas is needed at the compressor suction to obtain best efficiencies. (Refer to Chap. 18, for further data on dehydration.) This is true of all gases with low heat of vaporization, as particles of liquid on striking the cylinder are flashed into gas, restricting the entrance of further gas. Fig. 13 shows that at least 30 to 35 deg of superheat are needed to assure all liquid entrainment being eliminated. By using a heat exchanger, it is possible to obtain the total temperature at the compressor in the neighborhood of 65 F.

Heat Exchangers

25. The surface area required in a heat exchanger for a refrigerating system can be estimated by the following formula:

$$\frac{Q}{(T_1-T_s)25} \tag{8}$$

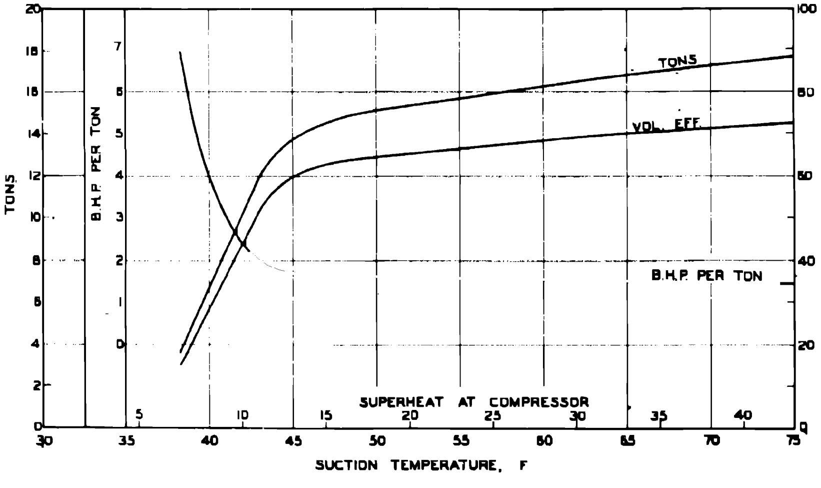


Fig. 13. Variable Superheat at Constant Pressure

(Refrigerant—Freon; suction pressure 31 psig; discharge pressure 140 psig; barometer 29 in at 80 F)

Where

Q = capacity of the condensing unit ×0.30, Btu per hr
T_1 = temperature of liquid entering the exchanger, F
T_s = temperature of suction gas entering the exchanger, F

Care is to be exercised that the velocity of gas through the exchanger is not less than 1,200 fpm. No low points or traps are desirable, as oil may collect and suddenly surge into the compressor. In systems of ¼ and ⅓ hp, soldering the suction and liquid lines together for some length makes an effective exchanger.[8]

Automatic Oil Separators

26. The purpose of an oil separator is to prevent oil from entering the evaporator. Oil separators are not generally used on small units as the compressor is designed to pump small percentages of oil with the refrigerant. The effects in the evaporator are shown in **Tables 5** and **6**. When the velocity of gases passing through a coil is more than 1,200 fpm, oil will usually return to the compressor crankcase and not greatly affect the capacity.

In flooded evaporators of large internal volume, velocity is far below this requirement. Oil is not returned to the compressor, but loads up the evaporator. In such applications, it is essential that an automatic oil separator be used.

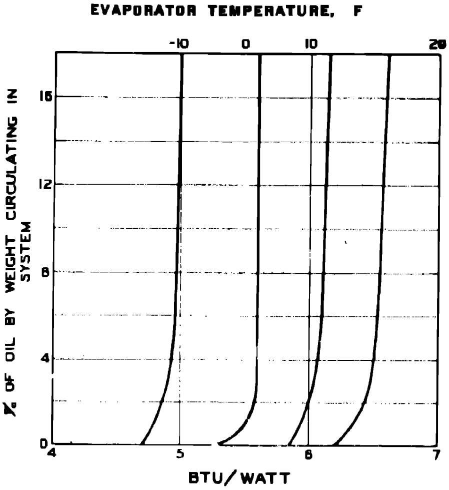


Fig. 14. Effect of Oil Circulation on Compressor Efficiency

When oil accumulates in a refrigeration system, the lower the refrigerant temperature, the greater effect it has on the evaporative capacity. This is shown in Table 6 by the rise in refrigerant temperature over that of pure Freon-12 refrigerant. Oil circulation in the compressor proper, due to better sealing of the suction and discharge valves, pistons and other moving parts, reduces kw input and increases the volumetric efficiency as indicated by referring to **Fig. 14.**[7]

Table 5. Capacity as Affected by Oil

Percentage of oil in Freon-12 by weight	Percentage reduction of capacity in evaporator due to oil mixture
0	0
1	½
2	1¼
3	2¼
4	3
5	4
6	4¾
7	5½
8	6¼
9	7
10	8
15	13
20	18½
30	32½
40	48
50	63

Table 6. Increase in Boiling Point Due to Oil in Freon-12, deg F

Boiling point of pure Freon-12, F	Part of oil in circulation by weight, %				
	10	20	30	40	50
0	1.50	3.50	5.50	8.0	11.50
5	1.36	3.3	5.3	7.8	11.2
10	1.2	3.12	5.15	7.6	10.87
15	1.05	3.0	5.0	7.36	10.53
20	.9	2.8	4.7	7.15	10.25
25	.75	2.6	4.65	6.9	9.9
30	.6	2.46	4.5	6.7	9.6
40	.3	2.12	4.15	6.25	9.0

Condensers

27. Air-cooled condensers are generally made of copper tube and copper fins, copper tube and steel fins, or steel tube and steel fins. Steel fins and tubes are recommended because the price is less than that of copper, and the likelihood of electrolysis is eliminated.

Fins are usually made of .006 to .015 in. steel or copper, sufficiently kinked for strength and generally set five to seven per inch, depending on the spacing of the tube and fin thickness.

The fins are forced on the tubes and properly bonded by dipping in a bath of lead or tin, to make a direct bond to the tube. The outside surface is either tinned or painted with some pigment paint, suitable for temperatures up to 250 F. Painting does not affect heat transfer.

Table 7 shows the Btu heat removal required per ton of refrigeration, when rated in accordance with the ASRE method using 65 deg suction vapor temperature.

The practice as to adequate condenser surface, when the primary and secondary surface is considered, is to use 150 sq ft of total surface per ton of refrigeration with the average air velocity of 500 fpm. This applies to full-length condensers, covering the whole condensing unit assembly. When a condenser is placed directly over the fan, proportionately less surface and higher air velocities are required, from 800 to 1,000 fpm. Higher air velocity increases the noise factor. The edge of the fan should be placed ¾ to 1¼ in. away from the condenser. The choice of fan is a problem of experimentation.

Condensers, when applied to low-pressure refrigerants, are used from −40 F to +50 F. Manufacturers have adopted and catalogued commercial condensing units as low-temperature applications, ranging from −40 F to 0 F; medium-temperature, −10 F to 25 F; and high-temperature, 30 F to 50 F.

28. Water-cooled condensers are generally classified into three types as follows:

a. **Shell-and-tube** type is similar to the familiar ammonia construction, with tubes rolled into tube sheets. Steel shells and tube sheets with copper tubes, usually finned, are common. For marine use, shells are copper with cupronickel tubes and tube sheets. The gases to be condensed are on the outside of the tube. The water travels in a multipass flow inside of the tubes. Heads can be removed to clean tubes when necessary; ⅝-in. OD copper tubes with .042-in. wall, finned, make a compact tube design.

b. The second type, the **double-pipe,** usually made of copper, is a tube within a tube, using the counterflow principle. The refrigerant passes from the head of the compressor between the outer and inner tubes, while water passes from the bottom up, within the inner tube. Tube sizes may be ¾-in. OD, .042-in. wall outside, and ½-in. OD, .035-in. wall inside; or 1-in. OD, .05-in. wall outside, and ¾-in. OD, .042-in. wall inside.

c. The third type, known as the **shell-and-coil** type, uses an outer shell and a water tube bundle usually finned, placed inside with inlet and outlet water nipple projections. This type of condenser is built with a primary surface of copper tubing,

Table 7. Heat Removed from Freon-12 by Condenser, Btu Per Ton of Refrigeration

(65 F suction vapor entering compressor)

Condensing temp, F		90	100	110	120
Condensing pressure, psig		99.6	116.9	136.0	157.1
Suction temp, F	Suction pressure, psig				
0	9.17	16,320	16,930	17,600	18,360
10	14.15	15,620	16,220	16,850	17,550
20	21.05	14,950	15,550	16,100	16,750
30	28.46	14,350	14,900	15,400	16,000
40	36.98	13,830	14,320	14,750	15,250
50	46.69	13,400	13,800	14,150	14,600

and has a secondary surface made of fins, usually spaced from four to seven fins per inch. The water tube bundle is made either as a helical coil or a nest of straight tubes suitably manifolded, inserted inside the shell and rigidly fastened in place. The shell is then closed and welded. Common tube sizes are ½-in. OD, ⅝-in. OD, or ¾-in. OD copper tubes. This arrangement does not permit cleaning of tubes except by chemical action.

The surface area required per ton of refrigeration for the shell-and-tube type is 12 sq ft, for the double-pipe type 9 sq ft, and for the shell-and-coil type 24 sq ft. These values are based on the capacity of the condensing unit at 50 F evaporator temperature, or at the peak temperature for the particular operating range.

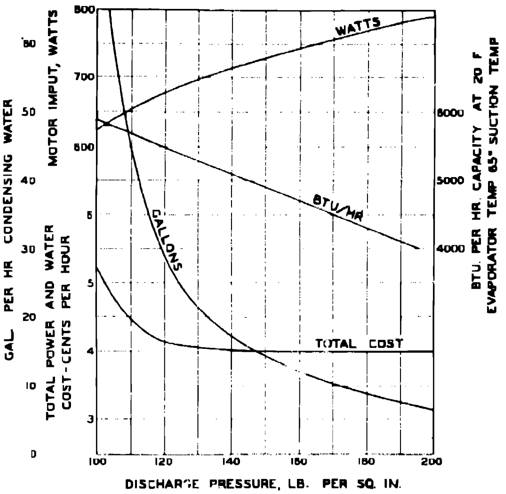


Fig. 15. Water-Cooled Freon-12, ½-hp Compressor; 2-cylinder, 1-13/16 in. bore × 1⅜ in. stroke, 580 rpm

There is a tendency to maintain low pressures on condensing units without considering the balance between cost of power and cost of water. **Fig. 15** shows the quantity of water and power required for a ½-hp Freon condensing unit. It is to be seen that with water at 25¢ per 1,000 gal or $1.87 per 1,000 cu ft and power at five cents per kilowatt, it is more economical to operate with head pressures from 130 to 150 psi.

Evaporative Condensers

29. An evaporative condenser for the small condensing unit may comprise only a coil with a propeller fan. Into the fan is directed a stream of water and no water is recirculated. Larger sizes provide fans drawing air up through the wetted condenser coil, while water is sprayed over the coil by a recirculating pump. The following formulas permit ready check of the evaporative condenser requirement for the small condensing units. The amount of air required is

$$\frac{\text{Condensing unit capacity, Btu/hr}}{3.5(h_2-h_1)} \qquad (9)$$

Where

h_2 = total heat in the leaving air, Btu per lb at a wet bulb temperature of T_c-15

T_c = refrigerant condensing temperature, F

h_1 = total heat in entering air, Btu per hr at atmospheric wet bulb temperature of T_A

T_A = wet bulb temperature, atmospheric air, F

The air required ranges between 250 and 300 cfm per ton. Surface requirements for finned tubing can be approximated by the following formula:

$$B_f=\frac{\text{Condensing unit capacity, Btu/hr}}{6(T_C-T_A+15)} \qquad (10)$$

Where B_f = surface area, sq ft, including prime and fin surface. (The ratio of fin to prime must not exceed 10 to 1.)

Surface requirements where all the surface is prime will be approximated as follows:

$$B_p=\frac{\text{Condensing unit capacity, Btu/hr}}{19(T_C-T_A+15)} \qquad (11)$$

Where B_p = area of prime surface, sq ft.

The size of the casing for induced air flow will be approximated by the following:

$$\frac{\text{Air required, cfm}}{400} \tag{12}$$

The amount of water circulated for the sprays will closely approximate 3.5 × area of casing. The amount of make-up water necessary will be:

$$\frac{\text{Condensing unit capacity, Btu/hr}}{300{,}000} \tag{13}$$

Gasoline Engine Drive

30. Whenever electric power is unavailable, such as on the farm, or for bus or truck transportation, an air-cooled gasoline engine is mounted on the condensing unit in place of the motor. The engine ratings should be at least 2.5 to 3.0 times the hp of the motor it replaces, to take care of overloads.

The engine should be a constant-speed type, controlled by a fly-ball governor connected to a butterfly valve on the carburetor. It is advisable to provide a clutch pulley on the engine drive, so that the engine can be cranked without turning the compressor.

Condensing units are usually furnished without controls and are started by hand. More elaborate installations for train and bus air conditioning applications provide automatic starting and stopping. Engines can be started by hand and stopped with a reverse-acting pressure or temperature control, with contacts to short circuit the magneto and stop the engine. On air-cooled condensing units, the gasoline engine should be arranged so that its flywheel fan blows air away from the condenser, for otherwise hot air from the engine will raise the head pressure. Some manufacturers prefer to use a jack shaft at the point where an electric motor would ordinarily be mounted, and arrange the gasoline engine on a separate base adjacent to the condensing unit.

The oil in air-cooled gasoline engines of small size should be changed every 50 hr. These engines consume lubricating oil at the rate of approximately 1 gal every 300 hr. Engine speeds vary from 1,800 to 3,600 rpm. Fuel consumption is about 1.2 lb per bhphr, or 1.66 pints per bhphr.

An interesting addition to the engine-driven condensing unit is the propane cycle refrigeration unit, developed by Schlumbohm.[7] In this system, gas from the condenser operates the engine. The quantity removed is automatically recharged into the system from a charging drum.

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32. COMMERCIAL LOW SIDES

AN EVAPORATOR is that part of a mechanical refrigeration system in which the liquid refrigerant is evaporated, and the heat required for evaporation is absorbed from the medium to be cooled. The medium cooled may be a gas such as air, a liquid such as water, or a solid such as packaged food.

Evaporator Materials and Refrigerants

1. Evaporators are usually made of copper, steel, brass, iron, or aluminum, depending upon the use of the evaporator and the refrigerant employed.

The most commonly used refrigerants in commercial installations are Freon-12, Freon-22, ammonia, methyl chloride, sulfur dioxide, and carbon dioxide. The Freons and sulfur dioxide can be employed with copper, steel, brass, iron, or aluminum. Methyl chloride may be used with any of these materials with the exception of aluminum or aluminum-base alloys. Ammonia can be used with iron, steel, or aluminum evaporators. Most any materials may be used with carbon dioxide.

Magnesium-base alloys should not be used for any of the Freons or for methyl chloride. Zinc and aluminum alloys should not be used with methyl chloride. Copper or its alloys must not be used with ammonia.

Refrigerant Control

2. Most small and moderate sized commercial installations employ automatic means of controlling the flow of refrigerant to the evaporator. The following methods of automatic flow control are used:

a. Dry expansion evaporators
    1. Thermostatic expansion valve
    2. Automatic expansion valve
    3. Capillary tube

b. Flooded evaporators
    1. Low-side float valve
    2. High-side float valve
    3. Injector nozzle

Details regarding the above methods of flow control are discussed in other chapters. The above controls are used to keep the evaporator operating efficiently when the system is calling for cooling. In addition, the following controls are also used to throttle or stop the flow of refrigerant when the cooling requirements are partially or totally satisfied:

a. Solenoid or magnetic stop valves usually in the liquid line.

b. Back pressure, suction pressure, or two-temperature valves in the suction line.

The compressor motor in the system is usually controlled by either a suction pressure cutout or by a thermostat located in or on the medium being cooled.

The combination of thermostatic expansion valve and dry expansion type evaporator usually is lower in cost, easier to construct to the required shape, less complicated to install, more dependable in returning oil to the compressor, and requires less space than a float-controlled flooded-type evaporator with its auxiliaries. For these reasons, the thermostatic expansion valve is the most commonly used control for commercial applications. In applications where the level of the evaporator is constantly shifting, as on boats, trucks, and trains, the flooded types will usually not give as satisfactory operation as the dry expansion type. The quantity of refrigerant required is much greater with the flooded type than with the dry expansion type.

Better heat transfer, however, can be obtained with liquid flowing through the entire evaporator as in a flooded type. This means that a smaller and lighter evaporator may be used than with expansion valve control.

In recent years there has been an increased use of the capillary tube as an expansion device, particularly in systems using hermetically sealed compressors of the fractional horsepower sizes. If there is no other flow restriction in the system except the compressor valves, the pressures in the system will equalize during shutdown and reduce the compressor starting load.

Defrosting

3. In this chapter, defrosting will be referred to as one of two types:

a. Natural
b. Artificial

Natural defrosting is that type which occurs during the idle periods of the system and which requires no automatic or manual attention. Its use is limited to evaporators in spaces having temperatures above freezing. Artificial defrosting is the type where the defrosting operation is performed either manually or by automatic controls. It includes scraping off the frost by hand, running water over the evaporator, piping the hot compressor discharge gas through the evaporator, or the application of some external heat.

A typical hot-gas defrosting hookup is shown in **Fig. 1.** During normal operation the valves A, B, E, and F are closed, and C and D are opened. In order to defrost the left evaporator, valve C is closed, and valves A and E are opened. The hot compressed gas then leaves the compressor and flows through the discharge lines to this evaporator through the valve A. The heat which is absorbed melts the frost, and the gas is condensed. The liquid then flows through the evaporator, the valve E, and the liquid lines to the expansion valve of the right evaporator. When the ice has been completely removed from the first evaporator, the evaporator is returned to normal operation by closing valves A and E, and opening C. The second evaporator is defrosted in a similar manner.

Types and Purposes

4. Evaporator construction may be generally classified as to three types.

a. Bare pipe or tubing is most generally applied where the air temperature to be maintained is below 34 F, and for liquid cooling where the coils are submerged.

b. Plate-surface evaporators usually consist of two pieces of flat sheet metal soldered or welded together at the outside edges, with either welded internal construction or formed tubing inside to provide refrigerant passage. Those with welded internal construction have the refrigerant in direct contact with sheet metal surfaces. The other type, in which the refrigerant passage is through formed tubing, usually has a vacuum or an eutectic solution in the space between the tubes. The vacuum permits atmospheric pressure to force the outside flat sheets against the tubing thereby providing a better, more positive heat transfer bond between them.

The direct refrigerant contact type and the vacuum type plates are used for shelf surfaces in freezer rooms and boxes and in

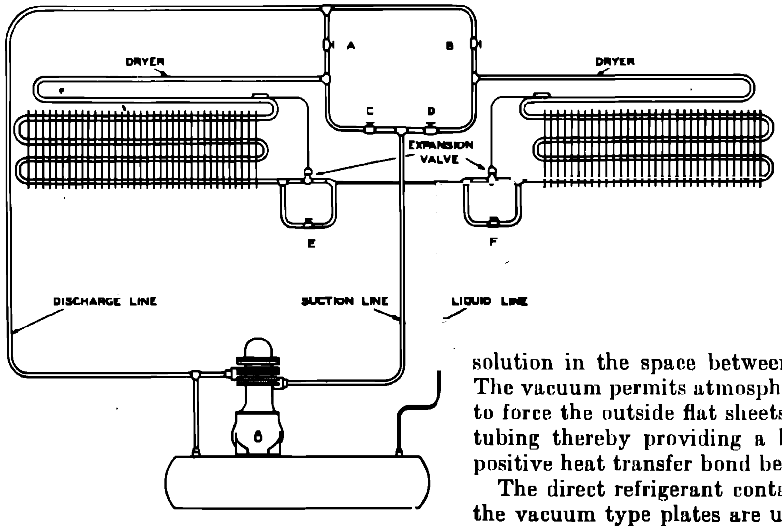

Fig. 1. Typical Hot Gas Defrosting Hookup

display cases in place of bare pipe, in low temperature (below 34 F) storage rooms, and in liquid chilling tanks. The type containing an eutectic solution is generally used in conjunction with a central refrigerating plant such as for refrigerated trucks, where the plates in the truck are connected to the refrigerating system during the night. The eutectic solution is frozen in order to supply sufficient hold-over refrigeration for the truck the following day.

Instead of refrigerated plates, sheet metal plates with refrigerant tubing soldered to the underside are sometimes used as salad or meat pans in various commercial cabinets and fixtures, and are used also in some freezer cabinets.

Baudelot-type coolers, sometimes referred to as aerators, are often used for milk cooling in dairies and water cooling in beverage plants. The evaporator for this unit is a vertical coil of tubing sometimes covered by stainless steel plates formed around the coil and in contact with it. In other designs, the plates are stamped to form the coil itself with the refrigerant flowing inside the corrugated passages. The unit is usually mounted in a vertical position with the corrugations horizontal. The liquid to be cooled is fed by gravity from a distributor across the top. The liquid flows down over the plate surface in a thin sheet. No damage is done should the liquid freeze at times. A refrigerant injector nozzle is often used. One disadvantage of this type of cooler is that the liquid being cooled is at atmospheric pressure, and is not in a closed system.

c. Finned-surface evaporators are generally limited in use to air-cooling applications where the temperature maintained is above 34 F, especially with respect to coils operating with gravity air-flow circulation. There are several types of forced-air coolers on the market using finned coils designed to operate in low temperature spaces. Most of these units have incorporated in their design some method of artificial defrosting. The fins are generally spaced 2 to 3 per in. instead of the more common spacing of 6 to 8 per in. used for above freezing temperatures. Finned coils, as well as bare pipe coils, are used in brine-spray units.

Bare Pipe and Flat Plate Surfaces for Air Cooling

5. a. Bare pipe and flat plate evaporators, while giving satisfactory operation at all temperatures, are generally limited to applications requiring 34 F and below. Bare pipe evaporators are usually constructed of scale-free pipe on the larger installations and copper tubing on the smaller jobs. While bare pipe evaporators can be made in practically any shape or size, they are usually found in one of three common designs (see **Fig. 2**). The most common type is the flat evaporator which is either hung on the walls or ceiling or sometimes used as a shelf. Where the space is limited, zig-zag pipe coils are often employed. For fixtures having overhead evaporator bunkers, the oval pipe coil is used.

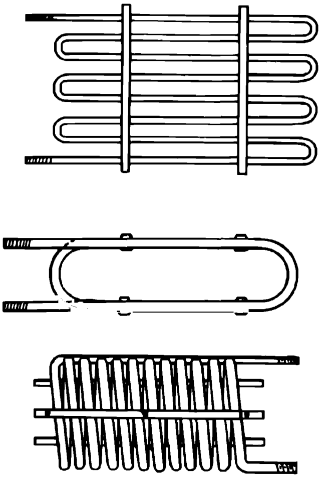

Fig. 2. Bare Pipe Coil Types

The flat plate type evaporator is often used in place of the wall or ceiling type pipe coil. For shelves where products are stored or frozen, the flat plate type is superior to the pipe coil due to the increased surface of the evaporator in contact with the product to be frozen.

A considerable thickness of frost can be accumulated on pipe and plate surface before the capacity is materially reduced. Periodic artificial defrosting, however, will be necessary, the frequency of defrosting depending upon the moisture available from the product, being greatest for open, moist products and least for canned products. Defrosting is accomplished either by hand scraping or by the application of heat.

b. Heat transfer values for pipe coils with gravity air circulation, as recommended by most manufacturers, vary from 1.6 to 2.3 Btu per sq ft hr F. **Table 1** indicates pipe capacities for 1-in. pipe. These capacities are based on heat transfer values closely approximating the above figures. The heat transfer of gravity air circulation pipe coils improves with greater temperature difference, due to the increased air circulation. Where the coils are used for shelves in contact with the product to be cooled, heat transfer will not only take place by convection but also by conduction. This will result in a higher rate of heat transfer, which is generally considered to be 15 to 25% greater than that for gravity air circulation coils.

Table 1. Pipe Coil Capacities, Gravity Air Circulation

Btu per hr per lin ft of 1-in. standard or extra heavy pipe)

Refrig temp, F	Fixture temperature, F				
	−20	0	20	30	40
25					11.1
20					16.2
15				9.3	21.0
10				12.7	25.8
5			7.5	16.3	30.4
0			11.0	19.9	34.9
− 5			14.6	23.5	
−10			18.4	27.5	
−15		7.5	21.6		
−20		11.0	24.8		
−25		14.6			
−30		18.3			
−35	7.3	21.5			

Most of the plate surface manufacturers give ratings from 2.0 to 2.5 Btu per hr sq ft F, considering both sides of the plate effective when used in "still air" at 40 F. These are based on frost-free plates. Values of 2.0 and less are given when used in 0 F air, assuming some frost accumulation. When the plates are used as shelf surfaces with the product to be chilled or frozen resting thereon, the heat transfer rate (to the product) is increased 15 to 25%.

c. Drier or superheating surface is recommended with all gravity air circulation pipe coils and plates controlled with thermostatic expansion valves. The drier surface consists of an additional length of pipe or plate surface. **Table 2** shows the added lengths sometimes used. When two spaces are cooled with one condensing unit and the bare pipe evaporators are connected in series the drier is added to the end of the last coil. In this case the drier coil obtained in Table 2 is usually increased by the amount shown in **Table 3.** Some manufacturers recommend that either the bare pipe evaporator be increased 25% to provide drier surface or a liquid-gas heat exchanger be used.

Table 2. Recommended Drier Pipe Lengths for Bare Pipe Coils

Fixture temperature, F	Length of pipe coil, ft				
	50	100	150	200	250
	Length added for drier, ft				
−20	20	33	46	56	64
−10	18	30	42	50	57
0	16	27	38	45	51
10	14	24	33	40	45
20	13	21	29	35	39
30	11	18	24	30	33
40	9	15	19	24	27
50	7	12	15	18	20
60	5	9	11	13	15

Table 3. Drier Pipe Coil Factors for Use with Evaporators in Series

Per cent of total load in last evaporator	Multiplier
20	2.1
30	1.9
40	1.7
50	1.5
60	1.3
70	1.1
80	1.0

d. The thermal bulb of the thermostatic expansion valve should be located on the condensing unit side of either the drier coil or the heat exchanger. With the low heat transfer rate obtained from gravity air circulation pipe coils, satisfactory opera-

Table 4. Maximum Load, Btu per hr in Freon-12 Evaporators for Each Thermostatic Expansion Valve*

Equivalent length of pipe or tube per valve, ft	OD size of tube, in.							Standard size of pipe or tube, in.						
	3/8	1/2	5/8	3/4	7/8	1	1 1/8	3/8	1/2	3/4	1	1 1/4	1 1/2	2
40	1,130													
50	1,010													
60	920	2,400						3,400						
70	850	2,200	4,300					3,120						
80	795	2,050	4,000					2,920	5,100					
90	755	1,950	3,800					2,750	4,850					
100	720	1,850	3,550	5,800	8,400			2,600	4,650	9,900				
110	685	1,770	3,410	5,550	8,000			2,500	4,400	9,500				
120	655	1,690	3,270	5,300	7,700			2,400	4,200	9,050				
130	625	1,620	3,120	5,100	7,400			2,300	4,050	8,700				
140	605	1,550	3,010	4,900	7,120			2,200	3,900	8,400				
150	585	1,500	2,900	4,750	6,850	11,800	13,100	2,150	3,780	8,100	14.600	30,300		
200	510	1,295	2,500	4,100	5,950	10,250	11,400	1,850	3,290	7,000	12,600	26,300	38,300	
250	455	1,150	2,225	3,650	5,300	9,200	10,100	1,650	2,900	6,250	11,300	23,500	34,200	63,000
300	415	1,050	2,030	3,320	4,900	8,350	9,200	1,500	2,670	5,700	10,300	21,400	31,400	57,500
350	385	965	1,900	3,075	4,500	7,700	8,500	1,390	2,480	5,250	9,500	19,900	29,000	53,000
400	360	910	1,770	2,860	4,250	7,250	7,950	1,290	2,300	4,930	8,900	18,500	27,100	49,800
450	340	850	1,670	2,700	4,000	6,850	7,500	1,220	2,170	4,620	8,350	17,500	25,600	47,000
500	320	810	1,500	2,500	3,700	6,400	7,100	1,150	2,050	4,400	7,900	16,500	24,400	44,500
550	310	770	1,490	2,430	3,550	6,100	6,800	1,100	1,970	4,175	7,500	15,700	23,200	42,500
600	295	740	1,430	2,320	3,400	5,900	6,500	1,045	1,900	4,000	7,200	15,100	22,100	40,500
650	285	705	1,370	2,220	3,300	5,700	6,250	1,010	1,830	3,850	6,980	14,500	21,300	39,000
700	272	680	1,320	2,150	3,200	5,450	6,000	970	1,750	3,700	6,700	14,000	20,500	37,500
750	263	650	1,270	2,080	3,100	5,300	5,800	940	1,700	3,550	6,550	13,500	19,800	36,500
800	225	630	1,230	2,015	2,990	5,100	5,620	910	1,650	3,460	6,250	13,100	19,200	35,200
850		610	1,200	1,950	2,900	4,990	5,450	880	1,600	3,350	6,100	12,600	18,600	34,200
900		596	1,160	1,905	2,800	4,850	5,300	850	1,550	3,280	5,900	12,300	18,100	33,200
950		580	1,135	1,850	2,750	4,725	5,150	830	1,510	3,180	5,750	12,000	17,650	32,500
1,000		570	1,120	1,820	2,700	4,600	5,050	810	1,475	3,100	5,600	11,700	17,200	31,350
1,100			1,070	1,730	2,550	4,400	4,850		1,400	2,950	5,300	11,200	16,400	30,000
1,200			1,015	1,650	2,450	4,200	4,620		1,340	2,800	5,075	10,700	15,700	28,800
1,300			970	1,580	2,350	4,030	4,450		1,290	2,700	4,900	10,300	15,100	27,700
1,400			940	1,525	2,260	3,900	4,300		1,240	2,600	4,700	9,900	14,600	26,500
1,500			905	1,480	2,190	3,750	4,150		1,200	2,500	4,550	9,600	14,200	25,800
1,600				1,430	2,110	3,625	4,000			2,430	4,400	9,250	13,700	25,000
1,700				1,380	2,060	3,525	3,900			2,370	4,290	9,000	13,300	24,200
1,800					2,000	3,450	3,780			2,290	4,150	8,800	12,900	23,600
1,900					1,960	3,350	3,680			2,230	4,050	8,500	12,600	22,900
2,000					1,900	3,260	3,580			2,170	3,950	8,250	12,200	22,300
2,100						3,180	3,490				3,850	8,100	11,900	21,700
2,200						3,100	3,400				3,725	7,900	11,650	21,200
2,300						3,020	3,330				3,650	7,750	11,400	20,800
2,400						2,960	3,250				3,580	7,600	11,200	20,400
2,500						2,910	3,200					7,450	11,000	20,000
2,600												7,300	10,800	19,700
2,700												7,150	10,600	19,300
2,800												7,050	10,400	19,000
2,900												5,900	10,200	18,700
3,000												6,800	10,000	18,300
3,100													9,820	17,900
3,200													9,700	17,600
3,300													9,550	17,400
3,400													9,400	17,200
3,500													9,250	16,900
3,600														16,700
3,700														16,500
3,800														16,300
3,900														16,100
4,000														15,800

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* Reference should be made to Tables 5 and 6, and the example shown on page 548.
These data are to be used only for determining the number of circuits for a tube or pipe evaporator coil.

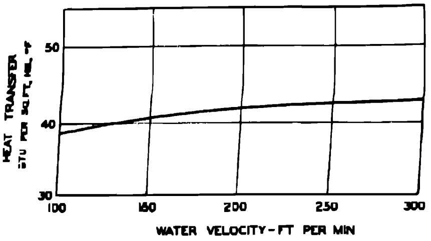


Fig. 3a. Heat Transfer of Bare Pipe or Copper Tubing Submerged in Water Bath, No Agitation of Bath, Water in Tubing

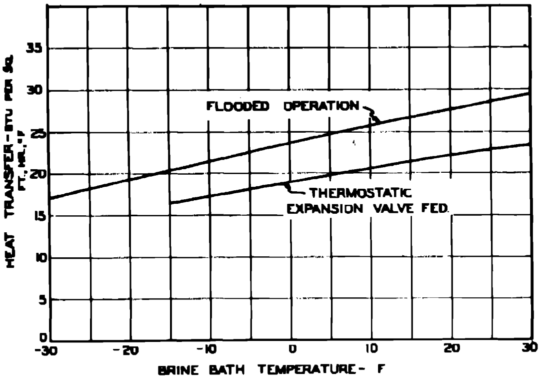


Fig. 3b. Heat Transfer of Bare Pipe or Copper Tubing Submerged in Brine Bath, No Agitation of Bath, Refrigerant in Tubing

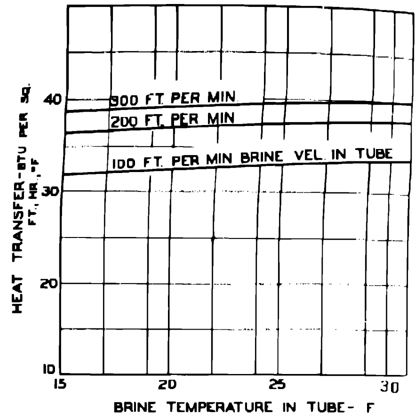


Fig. 3c. Heat Transfer of Bare Pipe or Copper Tubing Submerged in Water Bath, No Agitation of Bath, Brine in Tubing

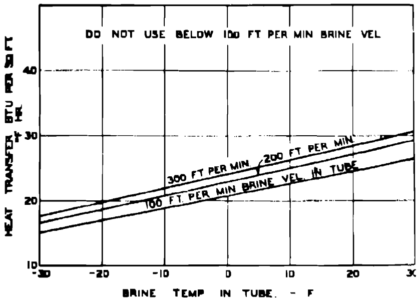


Fig. 3d. Heat Transfer of Bare Pipe or Copper Tubing Submerged in Brine Bath, No Agitation of Bath, Brine in Tubing

tion will be obtained with the thermal bulb on the condensing unit side of the heat exchanger.

Table 4 shows the recommended length of pipe or tubing per feed with various suction pressures and loadings. On installations requiring more than one feed controlled by thermostatic expansion valves, a thermostatic valve is usually put on each feed. Two or more lines can be fed with one valve, but quite often, this leads to refrigerant distribution problems.

Bare Pipe and Flat Plate Surfaces for Liquid Cooling

6. a. Submerged pipe coils are made chiefly in the flat or the oval shapes shown in Fig. 2, the cylindrical shape shown being used in small chambers.

b. The following **heat transfer** values for a non-agitated bath are obtainable when coils of the flat or oval shape are mounted vertically in the liquid bath for an approximate minimum height of two feet. Mounting the coils this way provides a satisfactory natural circulation of the bath. If the coils were mounted in a horizontal position, there would be little natural bath circulation and the rate of heat transfer would be reduced. This factor should be considered on those applications where it applies and a safety factor used. In a non-agitated water bath with thermostatic expansion control, the heat transfer for bare pipe or copper tubing is 30 Btu per sq ft hr F. When using this value, no drier surface is required. With a flooded evaporator and the other conditions the same as

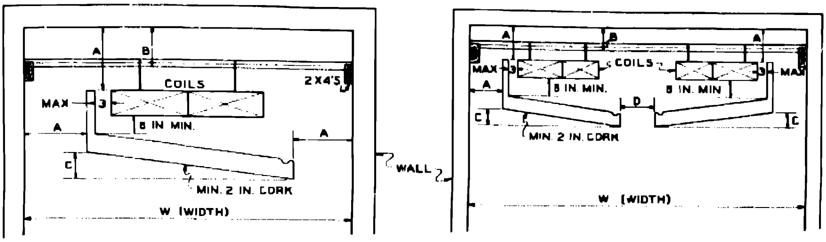


Fig. 4. Coil Arrangement for Single L Bunker (left); Coil Arrangement for Double L Bunker (right)

above, the heat transfer rate is 40. **Figs. 3a, 3b, 3c,** and **3d** indicate some recommended heat transfer rates for bare pipe or copper tubing in water; or brine baths with refrigerant, brine, or water in the pipe. For length of pipe or tubing per refrigerant feed see Table 4.

Table 5. Multiplier Table

Temp diff	Evaporator Temperature										
	−20	−15	−10	−5	0	5	10	15	20	25	30
10	.69	.77	.86	.96	1.06	1.18	1.31	1.45	1.59	1.74	1.90
15	.86	.97	1.08	1.20	1.32	1.46	1.61	1.78	1.95	2.13	2.32
20	1.00	1.12	1.23	1.37	1.51	1.68	1.87	2.06	2.27	2.49	2.71
25	1.14	1.28	1.42	1.57	1.73	1.90	2.10	2.31	2.55	2.81	3.08

Finned Surface with Gravity Air Circulation

7. a. Finned coils with gravity air circulation are employed in walk-in coolers, reach-in or grocery refrigerators, display cases, and counter cabinets. Although unit coolers have been developed and are partially replacing the gravity-type coils, for applications requiring minimum dehydration and loss of color, the gravity coil is usually recommended. For this reason gravity coils are ordinarily used in wholesale storage rooms and aging rooms. Gravity coils are also used where space or electric power limitations eliminate the use of forced air units.

Satisfactory coil and **baffle arrangements** are shown in **Figs. 4, 5,** and **6.** With the arrangements of Figs. 4 or 5, the coils

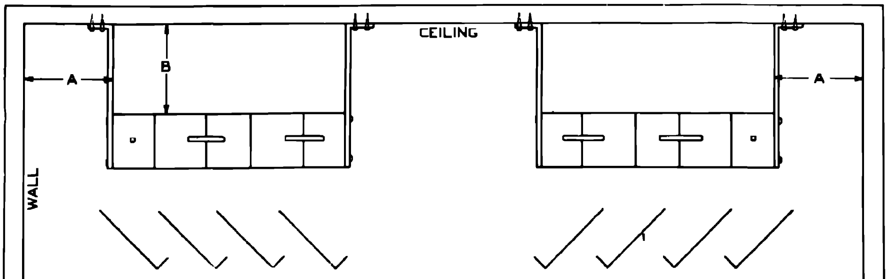
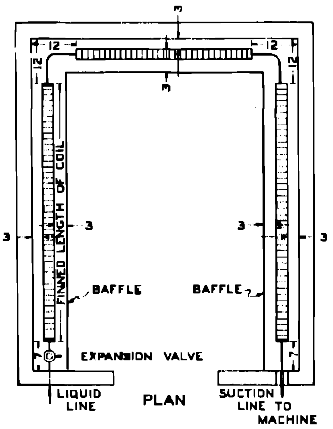


Fig. 5. Coil Arrangement with All-Metal Drain Baffles

Note—A equals 8 in. min, 12 in. max. B equals 6 in. max.

are one or two rows deep, that is, approximately 3 to 7 in. deep. Shallow coils spread over the ceiling are preferable for best air circulation over the products and for greatest unit surface efficiency. As the coil depth is increased, the air temperature range through the coil is increased, thereby reducing the mean temperatures, which finally results in less overall coil capacity.

The side wall coil of Fig. 6 delivers approximately the same heat transfer as the overhead coils due to the chimney effect of the baffles. These coils without the baffles will have about 50 to 60% of the capacity they will have when baffles are used.



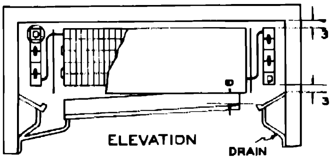


Fig. 6. Coils for Side Wall Installation

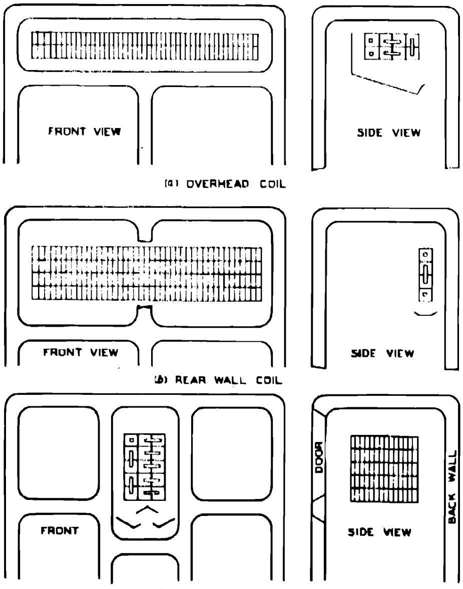
(Low ceiling storage rooms)

The effect of distance from ceiling for a one-row coil with five fins per inch without baffles has been studied. With a 40 deg temperature difference the coil capacity increased 33% in moving it from 1 in. to 3 in. below the ceiling. It increased an additional 17% when moved to 7 in. below the ceiling.

With arrangements of Figs. 4 and 5 the warm air flue should be located on the door side of the cooler in order that the warm air entering the door will pass through and drop its excess moisture on the coil before flowing over the products. Otherwise this moisture would condense on the cold products.

Several coil and baffle arrangements for grocery or reach-in refrigerators and display cases are shown in **Figs. 7** and **8**. In general, the coils and baffles are supplied and installed by the cabinet manufacturers.

In selecting finned-type evaporators the 24-hr daily load is first determined. A condensing unit is then selected on the basis that it will absorb this load in 12 to 16 hr of operation, in order to provide sufficient off periods for natural defrosting and excess capacity for peak loads. An evaporator is chosen which has a capacity equal to or slightly more than the condensing unit capacity. An evaporator too small will result in low relative humidity, product dehydration, greater temperature differences between refrigerant and air, difficulty in maintaining desired fixture temperature, longer running periods, and possibly a



(a) OVERHEAD COIL

(b) REAR WALL COIL

(c) CENTER BUNKER COIL

Fig. 7. Grocery or Reach-In Refrigerators

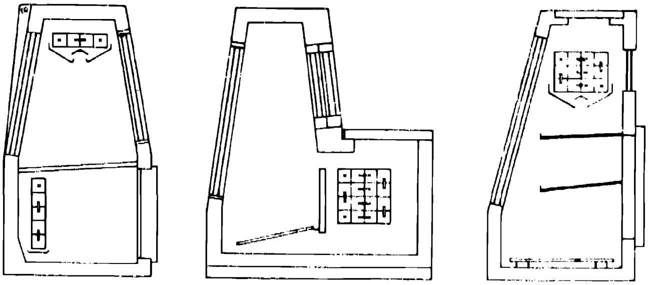


Fig. 8. Coil Arrangement for Display Cases

frost or ice problem. Too large an evaporator will reduce the operating time and the temperature difference, resulting in reduced air circulation, which may cause dead air pockets and a slimy condition of the products because of too high a relative humidity.

In general, satisfactory storage is obtained in retail store fixtures with a temperature difference between 15 and 25 F. For wholesale storage, usually for a longer period than retail service, the temperature difference can be broken down into three groups according to the minimum dehydration desired:

1. 15 to 20 F for cheese, cut meats, meat in quarters or halves, eggs, fresh fruit, fresh vegetables, poultry, and the like;
2. 20 to 25 F for meats, dried fruits, and vegetables, iced fish and beer in wood kegs;
3. Any temperature difference is satisfactory with products that cannot give up their moisture, such as canned or bottled goods and beer in steel kegs.

Table 6. Return Bend Table

Size of tubing or pipe, in.	Equiv ft of tubing or pipe per return bend
3/8	1.25
1/2	1.50
5/8	1.50
3/4	1.75
7/8	1.75
1	2.50
1 1/8	2.50
1 1/4	3.00
1 1/2	4.00
2	7.00

b. The rate of **heat transfer** for gravity air flow over direct expansion finned coils varies chiefly with the air velocity and air distribution over the coils. The velocity and distribution will vary considerably with each type of fixture depending upon the bunker, baffle and coil arrangement. It will also vary with the size of warm and cold air flues and the head of air pressure available for generating circulation as well as with the fin spacing, rows of tubes wide and deep, fin size and fin design of the coil. Hence it is difficult to offer specific heat transfer values that can be validly used in selecting the size evaporator required for a given load. Values can be taken from manufacturers' data obtained in accordance with the ASRE Standard 25-44.

Tests of transfer from air to finned surface have been made by Askin. Variables include the ice on the surface, variation in suction temperature, amount of superheat and the like. A coil was placed in a calorimeter with constant suction pressure maintained by means of an automatic expansion valve, refrigerant vapor being superheated a fixed amount. Fig. 9 shows the results, from which it was concluded that the greatest value is obtained in a coil that is as shallow as possible, with a wide fin pitch. In other words, it is better to use a 2-row, ½-in. pitch than to use a 4-row, ¼-in. pitch, with the same surface. Another factor is length; it is more economical to select a length of 80 in. and a width of 18 in. than a length of say 40 in. with a width of 36 in. It must be remembered that the heat transfer values of Fig. 9 are applicable only for the fixture tested and are included here only for the purpose of show-

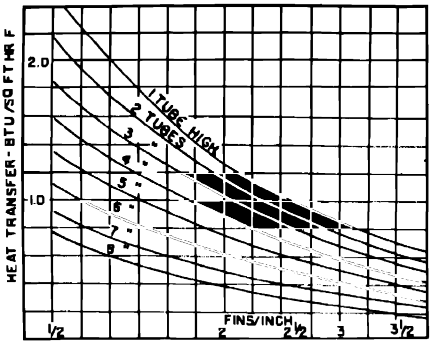


Fig. 9. Performance of Finned Coils under Natural Circulation (fins 24 in. wide × 2¾ in./tube high)

ing the effect of tubes high and fin spacing.

Generally speaking, the heat transfer of a specific type of surface will vary progressively from high to low in the following applications: (a) overhead and side wall coils in walk-in coolers; (b) overhead coils in a reach-in refrigerator; (c) side wall coils in reach-in refrigerators and display cases; (e) center bunker coils in a reach-in refrigerator and back bunker coils in display cases.

c. Drier coils are frequently recommended with gravity air flow finned surface, the drier coil consisting of a length of bare tubing located above the evaporator in the warm air. One manufacturer recommends 15 to 20 ft of drier coil, with the thermal bulb of the expansion valve located on the drier coil midway between the evaporator outlet and the point where the suction line leaves the fixture. The drier coil is recommended by this manufacturer for the purposes of providing superheating surface and for assurance that outside heat will not be conducted back along the suction line to the thermal bulb during the off cycle, otherwise the conducted heat would open the expansion valve and flood the evaporator.

On installations where the suction line must pass through spaces in which sweat dripping from the suction line would be objectionable, a liquid-gas heat exchanger located inside the fixture will reduce and quite often eliminate this problem. The use of the heat exchanger will also increase the capacity of the condensing unit due to the liquid subcooling obtained. Such a heat exchanger should also be used if the evaporator is located much above the condenser.

d. On most finned surface commercial applications the refrigerant flow is controlled by **thermostatic expansion valves.** From Table 4 the maximum recommended length of tubing for one refrigerant line can be determined in order to operate with a reasonable refrigerant pressure drop through the coil. In the event that more than one feed is required, it is considered preferable to use a thermostatic expansion valve with each feed. Two or more feeds can be controlled by one expansion valve, if the proper refrigerant distribution can be attained. There are multi-outlet expansion valves on the market that perform satisfactorily for use with properly designed multiple-feed evaporators.

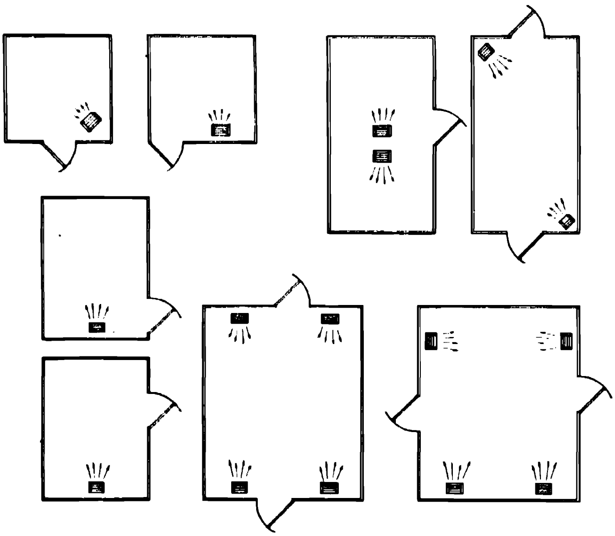


Fig. 10. Suggestions for Location of Unit Coolers in Walk-in Refrigerators

With finned surface on any evaporator operating with a comparatively high rate of heat transfer, the thermal bulb should be located on the evaporator side of liquid-gas heat exchangers for satisfactory operation, otherwise excessive pressure cycling will result.

Unit Coolers

8. a. During the past few years, unit coolers have gained wide acceptance for use in retail commercial refrigeration fixtures. From the many models and sizes on the market, a unit cooler can be selected for practically any application where gravity air flow finned coils are applicable, such as walk-in coolers, reach-in refrigerators, display cases and beverage cooling cabinets.

In general, these coolers are applied to maintain temperatures above 34 F, in order to operate on a natural defrosting cycle. There are, however, models that are designed for low-temperature applications, having included in their design a method for artificial defrosting. Three common methods of artificial defrosting are: (a) the use of hot discharge gas with the arrangement of Fig. 1; (b) city water distributed from a tube above the coil; and (c) electric strip heaters built into the unit, usually with adjustable panels for closing the supply and return air openings during defrosting.

From observation of the frost and ice formation of a unit operating in a 35 F space, it might appear that the ice would stop the air flow in a short time on a unit operating in a 0 F space. Actually this is not true, since the frost formed in a 35 F space is wet and sticks to the fins, while the frost formed in a 0 F space is snow-like in form, and a considerable quantity of it is blown off the surface.

There are many opinions as to the correct location of the unit in the cooled space for the most satisfactory performance. For walk-in coolers the recommendation of one manufacturer aims to draw the hot, moist air entering the door up through the coil in order to remove the excess moisture from the air with the coil instead of having this moisture condense upon the cold products. In selecting the location of the unit, sufficient space must be provided between the rear of the unit and the wall so that the air flow will not be restricted (Fig. 10).

Table 7. Recommended Maximum Temperature Difference and Air Velocity for Various Commodities

Commodity	Maximum temperature difference, F	Maximum air velocity, ft/min
Beef (quarters)	12	40
Beer (wooden kegs)	12	40
Berries	12	40
Butter	12	40
Cheese	12	below 25
Citrous Fruits	12	40
Cut Meats	12	below 25
Dried		
Fruits	16	60
Meats	16	60
Nuts	16	60
Vegetables	16	60
Eggs (crated)	12	40
Fish (iced)	16	60
Flowers (cut)	12	below 25
Fruits		
Dried	16	60
Fresh	12	40
Lamb (halves)	12	40
Lard	20	100
Malt	20	100
Maple sugar	20	100
Maple syrup	20	100
Meats		
Dried	16	60
Hams (fresh)	12	40
Loins (fresh)	12	40
Sausage (bulk)	12	below 25
Smoked	16	60
Nuts (dried)	16	60
Onions	16	60
Oysters	16	60
Pork (halves)	12	40
Poultry (fresh)	12	40
Sausages	12	40
Veal (halves)	12	below 65

In a specific fixture with fixed evaporator and fixed air temperatures, the dehydration of a moist product is considerably greater with a unit cooler than with gravity air flow finned surface, due, of course, to the more rapid air circulation with the unit cooler. For this reason most manufacturers list maximum permissible temperature differences for various products in their application literature **(Table 7)**.

Recommended minimum design evaporators vary between 20 to 25 F in spaces to be kept at about 34 F. Lower evaporator temperatures generally result in an icing problem.

b. The **heat transfer** rate of finned surface with forced air circulation is affected by so many variables that it is difficult to suggest any specific values that could be depended upon. Manufacturers' values based on unbiased tests made in accordance with the ASRE Standard 25-44 should be acceptable.

c. Drier coils are sometimes used to provide superheating surface. They will not eliminate suction line sweating, since the gas temperature can be raised only to the fixture temperature and in most instances this temperature is lower than the dew point of the outside space.

d. The refrigerant flow through the unit cooler is generally controlled by a **thermostatic expansion valve.** When liquid-gas heat exchangers are used, the thermal bulb of the thermostatic expansion valve should be located on the evaporator side of the heat exchanger to reduce pressure cycling to a minimum. The heat exchanger does not increase the efficiency of the evaporator, but does increase the condensing unit capacity due to liquid sub-cooling. Also, the liquid in most cases will raise the suction gas temperature leaving the fixture to the point where the line temperature will be above the dew point of the outside space and little or no sweating will occur.

Most engineers recommend that the unit cooler fan be operated continuously in order to defrost more rapidly during the off cycle.

Brine Spray Units

9. Brine spray units are used for many applications where coil temperatures below freezing are required. These units use either bare pipe or finned type coils. They usually are floor mounted. A pump draws the brine from the drain pan and forces it through nozzles over the coil. These units, built much like evaporative condensers, have replaced pipe coils and bunker spray units in packing houses, breweries and other installations. When using them it must be remembered that sodium brines have relatively high freezing temperatures. Calcium brines have lower freezing temperatures but it is not desirable to use calcium brine units for spaces containing exposed food products.

Example of Use of Tables in Design

Assume a storage room with a load of 7 tons at −10 F evaporator and 10 F room temperature, no heat exchanger being used; calculate the following using 2-in. pipe in 16-ft lengths with an overall heat transfer factor (U) of 1.6 Btu per sq ft, hr, deg F.

1. *Required length of pipe.* The area required is $(7 \times 12{,}000)/(1.6 \times 20) = 2{,}625$ sq ft. The linear feet per sq ft being 1.61, this calls for 4,226 ft, to which an allowance of 25% may be added for safety, making 5,280 ft.

2. *Length allowing for the return bends.* This amount of pipe in 16-ft lengths will total to $5{,}280/16 = 330$ passes. For 2-in. pipe, each return bend by Table 6 equals 7 ft, making a total equivalent length of pipe of 2,310. The total equivalent length is now $5{,}280 + 2{,}310 = 7{,}590$.

3. *Design with three expansion valves.* The equivalent length per feed in this case will be $7{,}590/3 = 2{,}530$ ft, the load per valve amounting to 28,000 Btu per hr. In Table 4 it is found that the maximum allowance is 19,900 ft, subject, however, to the multiplier of Table 5 which for the conditions stated is 1.23. Thus the allowable load is $19{,}900 \times 1.23 = 24{,}500$. The actual being greater, it appears that more than three expansion valves will be needed.

4. *Design with four expansion valves.* In this case the equivalent length per feed is 1,897 and the load 21,000 Btu per hr. Table 4 shows that 22,900 is permissible while the multiplier is still 1.23, making a total of 28,200, or more than the actual.

The minimum length per feed is to be based on the minimum gas velocity from the valve of 400 fpm, with the exception of $\frac{3}{8}$ and $\frac{1}{2}$-in. OD, where it should be 200 fpm.

33. COMMERCIAL REFRIGERATORS

LARGE refrigerators may be divided into eight classes: restaurant, large home, grocery, bottle, bakery, biological, frozen food and florist.

Restaurant Refrigerators

1. Restaurant refrigerators are built in 15, 20, 30, 40, 60 and 80-cu ft sizes. Because many of the smaller sizes are used behind counters the depth must be limited to about 28 to 30 in., depth of larger sizes being 33 to 34 in. The popular height is 72 in., allowing accessibility on the top shelf as well as on the lower shelf.

The **interior** of metal refrigerators should be lined with acid-resisting porcelain as it is easily cleaned and not affected by moisture or food. The exterior, where exposed to handling or rubbing, especially the doors and corners, should be made of porcelain, stainless steel, or other material not easily affected by abrasion.

Many restaurant men prefer **wood construction** refrigerators. The interiors are usually spruce-lined, sanded smooth and finished with clear odorless alcohol shellac, and the exterior is golden oak. Hardware should be rustproof, and of the self-closing type. Some restaurants require fasteners that can be padlocked, and most modern fasteners have this feature. Reach-in doors should be equipped with grease-proof gaskets and fitted to make an air-tight seal.

In all wood refrigerators, wood slat shelves are used, and in metal refrigerators porcelain slat or wire shelves are used. These have an adjustable feature so that they can be raised or lowered, giving flexibility to the refrigerator. An automatic light that turns on when the doors are opened is a very convenient feature on modern refrigerators.

The **coiling** should be designed to maintain temperatures of 36 to 42 F and the circulation should be designed to prevent interchange of odors. A fairly high humidity (70 to 85%) is required for all cut and uncovered foods to avoid excessive dehydration.

Gravity coiling is fast giving way to **forced air coiling.** Some manufacturers use a panel-type coil on the back wall, drawing warm air in at the top front and forcing it out at the bottom. Other manufacturers use a coil behind the door mullion which takes the warm air in at the top and forces it out at the bottom, and still others install a ceiling type unit in the center of the ceiling. This unit draws the air in at the center and forces it out the sides. The most satisfactory and successful unit is one that does not blow directly on the products in the refrigerator, but gives uniform circulation and is not greatly affected by service.

In view of the fact that restaurant refrigerators usually go in a hot kitchen, may stand next to a hot cook stove and are subjected to heavy service, they should be designed to operate satisfactorily in an ambient as high as 120 F.

Large Home Refrigerators

2. Refrigerators for large homes are usually of the restaurant type, depending upon the requirements. Many large homes require a refrigerator larger than the domestic type, but having the same general arrangement of ice-freezing trays, frozen-food storage capacity, and a large high-temperature storage compartment.

The best refrigeration equipment for some of the larger homes is the combination of a restaurant-type refrigerator for high-temperature storage and a frozen-food refrigerator or chest for ice-making and low-temperature storage.

Grocery Refrigerators

3. The general construction and sizes of grocery refrigerators are about the same as those of restaurant refrigerators. They differ in that they are equipped with glass doors. Another type of grocery refrigerator has sliding sash on the upper front and

storage doors in the bottom. It is brilliantly lighted with fluorescent lights, and because of the sliding doors is frequently used in self-service markets.

Products usually displayed in grocery refrigerators include milk, butter, cheese, eggs, lard, oleomargarine, yeast, soft drinks, etc. The milk compartment should be maintained below 40 F, whereas temperatures up to 45 F are satisfactory for other items. Cut items require a high humidity and a gentle circulation.

Heat loads for grocery refrigerators are usually figured for a 90 F ambient, and the machine specified to operate an average of 16 hr per day in an average ambient temperature of 90 F.

Bottle Refrigerators

4. Bottle refrigerators are used extensively where beer is sold in bottles or cases. Grocery-type refrigerators are often used as bottle refrigerators. However, when this is done, it is necessary that the coiling and machine be specified for bottle loads. The bottle refrigerator has these advantages over other types of bottle storage: (1) it will hold a large quantity of bottles; (2) the bottles are easily accessible on all shelves; and (3) if glass doors are used, they make a good display.

The heat load and machine specifications for bottle refrigerators will depend upon the customer's requirements. If only one load of bottles is to be cooled every 24 hr, the machine should take care of the refrigerator heat gain, including service, and at the same time cool the bottles. If the refrigerator is to be refilled two or three times a day, the machine and coils should be selected to fit these requirements. Since humidity is not a factor in bottle cooling, a greater temperature difference can be used than in restaurant refrigerators, in many cases making it unnecessary to redesign the coil for bottle cooling. Forced air coils are fastest and most efficient for this application due to positive, continuous circulation of the cooled air.

Bakery Refrigerators

5. It has only been in recent years that bakers have recognized the value of refrigeration for retarding dough as well as for keeping ingredients. The bakery refrigerator may be two or three compartments wide, or multiples of these. General construction follows closely the design and construction of restaurant refrigerators. The compartments for retarding doughs are equipped with slides for supporting the trays, the average tray being 18 × 26 in.

Coiling in bakery refrigerators is of the utmost importance. Coils must be designed to maintain humidities approaching 100%, to take heat out of the dough quickly, and maintain an average temperature of 36 F. In addition, the circulation of air must not be directed on the dough.

The capacity of the condensing unit required to cool the bakery refrigerator will depend upon the room temperature, size of refrigerator, amount of dough to be retarded, and service on the refrigerator.

Biological Refrigerators

6. Biological refrigerators are used for the refrigeration and storage of biological supplies, serums, etc. The general size and construction are similar to those of restaurant refrigerators except that instead of shelving drawers are used for storing the supplies. This helps in segregating the different supplies and at the same time increases accessibility. Average temperatures of 35 to 36 F are required. The humidity should be held below 50% to prevent labels coming off the bottles and packages. A thermostatic control is desirable and the bottoms of the drawers should be perforated to help maintain uniform temperatures. The down draft type of coil is quite satisfactory for this usage.

Because the preservation of biological supplies is of the utmost importance and refrigeration must be maintained under all conditions of ambient temperature, the machine should be specified to give satisfactory refrigeration in an ambient temperature as high as 120 F.

Frozen-Food Refrigerators

7. Frozen-food refrigerators are newcomers to the commercial refrigeration market. They are so new, in fact, that there are no standard sizes. Greater care must be taken in the design and construction of these refrigerators than with higher temperature refrigerators. They may be wood exterior and interior, but it is likely they will follow the trend of the higher temperature refrigerators and be metal exterior and interior.

The **insulation** should be at least 4 or 5 in. thick, depending on the type used, and must be sealed vapor-tight to prevent infiltration of moisture. The coiling may be built into the liner, but the popular arrangement is to use plate coil shelves which serve two purposes.

Since cold air will spill out like water when the door is opened on an upright type refrigerator, means must be taken to prevent this by the use of **inner doors** made of metal, glass, plastic, fibreboard, or drawers. Great care must be taken that the refrigerator door seals tightly all around. It should be equipped with freeze-proof type gasket. The hinges and fastener must be heavy enough to prevent sagging, and the insulation of the door must be sealed the same as in the rest of the cabinet.

The condensing unit should be selected to maintain an average temperature of 0 F in the refrigerated compartment with not more than 16 hr running time per day in a 90 F room. If there will be a freezing load, it must be considered when specifying the condensing unit.

Frozen-food chests. The simplest form of frozen-food container is a chest with lift lid. The construction should be similar to that of the upright type frozen-food refrigerator. Inner doors are not necessary, as the cold air will stay in the well, a definite advantage which saves refrigeration and simplifies the design. The disadvantage is that food stored in the lower part is not easily accessible.

Here again the coiling may be built into the liner or it may be made up of plate coil dividers. Each arrangement has its advantages. Proponents of the liner-type coiling claim that the coiling picks up the heat before it can get to the stored product and also that it allows more flexibility of the interior arrangement than do stationary plates. On the other hand, the liner coil increases the temperature difference between the two surfaces of the insulation, resulting in greater heat leak.

The condensing unit for this application should also be selected to run not more than 16 out of 24 hr in a 90 deg room, and if there is a freezing load it should be considered when specifying the condensing unit.

Florist Refrigerators

8. Basically, florist refrigeration is divided into three classes: display; storage of flowers, greens, and materials ready for use; and storage of bulbs and plants. No retail florist should be without a refrigerated, well-lighted display.

Since the flowers are sold from the display refrigerator, accessibility is important. The doors must open easily and be so arranged that flowers can be put in and taken out of the refrigerator quickly, and with the least disturbance to other stock already in it.

Many florists require both **display and storage,** in which case the display refrigerator is installed in the front part of the shop, and a walk-in cooler installed in the workshop in the rear. The storage cooler should be arranged with shelving and equipped with high humidity coils. It can be used for retarding bulbs, for extra stock, finished pieces, and miscellaneous items.

The wholesaler of florist stock may require both display and blind storage, depending upon his operation. For blind storage, walk-in coolers equipped with shelves and high humidity coils are recommended. The construction may be wood or metal. In the case of wood construction, care should be taken to prevent swelling and warpage due to high humidities inside the refrigerator. Florist display refrigerators should have 3-in. insulation in the walls and three thicknesses of glass in the display windows.

If gravity coils are used, they must be designed to maintain high humidity conditions. This is accomplished by operating them at a high back pressure and consequently a high coil temperature. At the same time, they must be baffled so that the intensity of the circulation does not strike the petals. Certain flowers such as gardenias will not stand circulation of air and must be kept in an enclosed space. The favored method is to keep them in a small glass or cellophane box on the shelf inside the refrigerator.

Forced air coils should preferably be of the panel type, and designed to maintain a high humidity and not blow on the flowers.

Certain precautions are necessary to

prevent damage by the refrigeration itself If a pressure switch is used for control, a small leak or valve freeze-up could cause the machine to run continuously and freeze the stock.

To guard against **damage by freezing,** the florist refrigerator should be equipped with a thermostat which automatically turns off the machine when the temperature for which it is set is reached. Refrigerants such as sulfur dioxide and ammonia are detrimental to flowers and should be avoided where they are likely to come in contact with the flowers should a leak occur.

Until recently temperatures of 45 to 50 F were considered satisfactory for florists' stocks. It has been demonstrated, however, that some flowers hold better at 40 to to 45 F and that temperatures approaching 32 F are desirable for retarding bulbs.

Machines for florist refrigerators should be of the high back-pressure type, and specified to operate approximately 50% of the time in a 90 F room.

Walk-in Coolers

The refrigerated walk-in cooler originated more than fifty years ago. Wood coolers with cork insulation are still popular and are difficult to improve upon for all-round efficiency, although white porcelain, aluminum, stainless steel, or Dulux exteriors and lighter but efficient insulations are now widely used.

Most coolers are built in sections so they may be easily transported and installed. The present trend is toward metal exterior and interior coolers built up of standard small sections and finished in Dulux or other suitable finish. This type of construction allows for efficient production, easier handling, easier cleaning and more flexibility of sizes.

Standard size coolers have been 4 × 6, 5 × 6, 7 × 5, 8 × 8, 10 × 8, 10 × 10 and 10 × 12 ft. With the new trend the depth dimension is limited only by the length of the top and floor sections, and the width may be any desired dimension as long as it is a multiple of the standard sections.

Three inches of insulation for wood construction coolers and five for metal construction coolers have been generally accepted to be the most efficient for general storage temperatures of 35 to 40 F, considering insulating efficiency and maximum storage space. The height depends upon the coiling and the wall thickness. The clear height of the cooler inside should be at least 6 ft so that tall men can walk in without stooping over.

The hardware on the doors should be of heavy non-rusting metal and of the self-closing type. The fastener on the entrance door should be arranged so that it can be easily opened from the inside. Some coolers are equipped with a bull's-eye light with switch on the outside door panel controlling the light on the inside of the coolers.

Meat Coolers

9. In meat markets reach-in doors with glass display are popular, both for their display value and facility. The entrance door may be on the end or in the front.

Shelving is placed behind the reach-in doors for the storing of lard, butter, eggs, and smaller pieces of meat. The walls of the cooler are equipped with meat rails, usually two per wall, for hanging halves and quarters, or with shelving.

The most successful coiling provides a constant, gentle circulation of air that does not blow on the meat with intensity. At the same time a relatively high humidity and a small temperature differential between cut-in and cutoff of the machine should be maintained. The coiling should take care of maximum load requirements, yet it should not cause excessive humidity and consequent slime during the off cycle. Coolers with overhead coiling are made 8½ to 9 ft high and are equipped with suitable louvered drain pans properly insulated to prevent condensation. New modern coolers are being built with forced air circulation, some with ceiling type units in the center of the cooler ceiling, and others with a down draft type unit on the cooler wall.

Vegetable Coolers

10. Vegetable cooler design and construction are essentially the same as for meat coolers, the main difference being that here the shelving takes the place of meat rails. The floor should be metal covered, preferably with galvanized iron, to prevent slipping. Floor racks over the metal floor are necessary to prevent slipping.

In the past vegetable coolers have been held at higher temperatures than meat coolers, a satisfactory procedure for short-time storage. However, it has been proved that temperatures ranging from 34 to 40 F are more suitable for most vegetables, especially if they are to be stored for more than several days.

Both gravity and forced air coiling are satisfactory. The coils must be selected to maintain a high relative humidity, and in the case of the forced air coil, the intensity of circulation should not be directed on the vegetables. If large quantities of warm vegetables are placed in the cooler at one time this extra load should be considered when specifying the coil and machine equipment.

Beer Coolers

11. Frequently walk-in coolers are used for cooling bottled and keg beer. Coolers of this type are known as blind storage coolers as they have no windows or small service doors. They may be of wood construction or of the new metal construction, both inside and out, with three inches of insulation and equipped with large capacity forced air coils for quick cooling. Since air velocity, direction of circulation and humidity are not factors, forced air coils blowing the circulation on the bottles or kegs are quite efficient. Some coolers in which a faucet and drip catcher are attached to the wall are also used as dispensers, and the beer is drawn directly from the kegs in the cooler.

If keg beer is kept in the cooler, normal heat load requirements are satisfactory. However, if bottled beer is kept in the cooler, the additional load of the number of bottles stored per day must be considered.

Restaurant Coolers

12. Walk-in coolers are a necessity in large restaurants, hotels and institutions. A restaurant of normal size has need for only one cooler for storing meats, vegetables, and dairy products. Larger restaurants, hotels, and institutions should have a three-compartment cooler of sufficient size to store meats in one compartment, vegetables in another, and dairy products in the third. The general construction is the same as for meat coolers but without display windows.

Wood exterior and interior, porcelain exterior and wood interior, and metal exterior and interior coolers are used, although many restaurant men still prefer the wood exterior coolers. In general, the construction is the same as for meat coolers. Since these coolers will be installed in or near the kitchen, which may be very hot and will be subjected to heavy service, the heat load requirements are greater than for other types of coolers. The machine and coils should be selected to maintain proper temperature in the cooler in ambient temperatures as high as 120 F.

Frozen-Food Coolers

13. The increasing popularity of frozen foods has created a need for a low-temperature cooler suitable for storing quantities of frozen foods. This cooler is necessarily of heavier construction, at least 6 in. of insulation being desirable. Great care must be taken to seal it with hydrolene, vapor-proof paper, or other suitable sealing material against the penetration of moisture vapor. The joints must be designed to make an air-tight fit and at the same time minimize conduction of heat through the framing.

The entrance door must be designed so that it will seal perfectly without danger of freezing shut. To accomplish this some manufacturers are using an outside overlap door. The hinges and hardware must be extra heavy to carry the load of the door.

Wood floor racks are required to allow circulation under the packages piled on the floor and at the same time prevent slipping.

Cooling may be accomplished with plate coil stands, overhead plate banks, plates on the side walls, water defrost blower coils, or hot gas defrost blower coils.

If blower coils of either the water defrost or hot gas defrost are used, they should be specified as not more than 10 deg temperature difference to prevent excessive accumulation of frost on the coils.

If there will be a freezing load, this should be added to the heat load of the cooler when specifying the machine equipment. The condensing unit should be specified to have sufficient capacity at the suction pressure at which it will operate.

Unless the condensing unit has a built-in separator, it should be equipped with one.

Bakery Coolers

14. Bakers are fast learning the benefit of dough retarding, and are making use of it on a larger scale. Many of them started out with a small dough-retarding refrigerator, and now require a larger bakery cooler, with an entrance door and ramp on one side for running racks and troughs into the cooler, and a row of reach-in doors and bakery slides on the other side for retarded dough. While the design of this cooler is different, the general construction is the same as that of the meat coolers.

The coiling should be of a type that does not direct the intensity of the air on the surface of the dough, but will maintain temperatures above 32 F and below 36 F, will maintain the humidity above 85%, and will take the heat out of the dough quickly.

Chill Bars

15. Chill bars are counter-type beverage dispensers used for cooling and dispensing bottled beer or other beverages. They come in various lengths, the most popular being 6, 8, 10, and 12 ft. The average height is 36 to 38 in. so that they will slip easily under a standard bar. Unless installed under a bar, they are usually equipped with Formica or similar acid-resistant and burn-proof top. The exterior may be Formica, metal or wood and the interior rust-proof metal. The back is equipped with sliding sash set at an angle to provide easy access to the interior.

Most chill bars are of the dry refrigeration type in which bottled beverages are cooled by plate or forced air coils. Forced air cooling permits easier cleaning and provides more flexibility in the arrangement of the bottles. Since many states restrict sales of alcoholic beverages to restaurants, a water-cooling attachment should be optional.

Refrigerated Display Cases

Refrigerated display cases are used for displaying meats, vegetables, delicatessen items, dairy products, candy, and frozen foods, each requiring its own special design and arrangement.

There has been a revolutionary trend in display case design during the past six or eight years. There are now available two distinct types of display cases—the closed and the open self-service type.

Closed-Type Meat Display Cases

16. In general, there are three types of closed-type refrigerated meat display cases; the **single-duty type** which has a display compartment only, the **double-duty type** which has a display and a storage compartment, and the **super-market case** which is similar to the single-duty type but is deeper and provides more display.

Display case lengths run from 4 to 18 ft but the standard or popular lengths are 5, 6, 8, 10, 12, 14 and 16 ft. The standard height for meat cases is 50 in. and the standard depth 34 in., with the exception of super-market cases which are 36 to 42 in. in depth.

The **contour** of the case is important. Consideration must be given to good visibility of the product displayed, whether it be on the main shelf or mezzanine shelf, and at the same time the slope of the front glass must be such that undesirable reflections will be at a minimum. Accessibility for the butcher is also important. The drain arrangement should be such as to facilitate cleaning of the case and cleaning of the trap.

Among the first problems that confront the design engineer are the best **wall thickness** and **thickness of insulation.** The thicker the wall, the less will be the heat flow. However, there is a practical minimum, for shelf space is a vital factor. Thickness of insulation is also important for cases with gravity circulation. Sufficient heat must leak through the walls, when there is no service on the case, to cause the machine to cycle frequently enough to maintain circulation and prevent the meat from sliming in normal temperatures. On the other hand, too much leak will raise operation costs. Three inches has been found to be the most efficient thickness of insulation, when available shelf space, preservation of stock, and cost of insulation in heat units per dollar are all considered.

The number of **panes of glass** to use in the front of the case is also important to the design engineer. The effectiveness of the display is reduced by the amount that the light is reduced in intensity or distorted in color in passing through the panes of glass. Each polished surface disperses 4.6% of the light passing through it, so there is a definite reason for using as few panes of glass as economy and good refrigeration will permit. Since glass is not a good insulator itself, dependence for insulation must be placed upon the air spaces between the panes. It is generally accepted, and tests have proved, that an air space of ½ to ¾ in. gives maximum insulating value without permitting circulation between the glasses. As indicated by **Fig. 1,** a design employing three panes of glass allows good visibility and is economical.

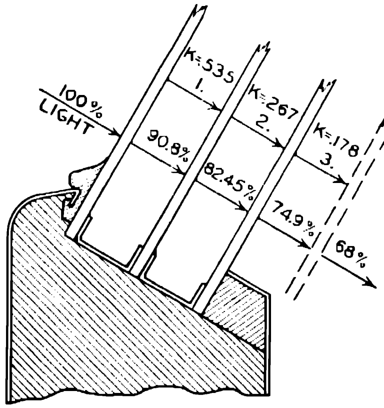


Fig. 1

According to the modern trend in lighting, display cases are supplied with **fluorescent lights.** Since fluorescent lamps give off much more light at a fraction of the wattage input taken by incandescent lamps, it is possible to install them inside the display fixture, a procedure which allows more efficient illumination and does away with undesirable reflections. Cases with mezzanine shelves cast a dark shadow on the main shelf when the top lights are turned on, so most manufacturers supply lights to be fitted underneath the mezzanine shelf. Care should be taken that heat from these lamps does not interfere with the circulation inside the case. Both hot cathode (low voltage) lamps and cold cathode (high voltage) lamps are used.

Glazing is an important construction detail, for if the front glasses become fogged, the display value of the fixture is nullified. It is difficult to seal the front glasses so that fog will not collect between them. Two methods of doing this are: (1) to seal the glasses tightly; and (2) to leave an opening to the inside of the case at the top and bottom, to allow breathing to the inside of the case. Because of its polished surface and freedom from imperfections, ¼-in. plate glass has been used extensively. However, some manufacturers use one sheet of ¼-in. plate glass on the outside and two sheets of $\frac{3}{16}$-in. sheet glass on the inside. Hard rubber sash glazed with two or three thicknesses of double-strength glass are generally used on the back of the case.

The interior of the case should be constructed of porcelain enamel on steel, stainless steel, or other material that can be washed easily and will not scratch, stain, or corrode. The drain trap and drain pipe must be sealed to prevent leakage of moisture into the insulation.

Most double-duty display cases are cooled with direct-expansion type finned coils in the top and in storage compartment. Single-duty types use bare tube or plate coils underneath the trays on the main shelf.

In the meat display cases employing **gravity circulation** with a top coil it is important to have a drip pan underneath the top coil, not only to catch the defrost water from the coil but also to guide the circulation. The efficiency of the entire system depends greatly upon how well the drip pan allows the warm air to circulate through the coil. In modern cases this drip pan is split or louvered, to allow free passage of air from the top coil. The drip pan must also be insulated to prevent condensation on the under side when the doors are opened. Most drip pans employ an air space of ⅜ to ½ in. between two thicknesses of metal having a moisture-resistant finish on the bottom surface.

Some case manufacturers use **forced air circulation** in which finned coils are installed in the bottom of the case and air is forced upward through the coils to the top of the case and then back to the fan through ductwork. Another method is to install a blower coil at one end of the case.

Air is drawn in along the horizontal length of the case in the top compartment and dispersed into the bottom compartment. Still another method is to install a blower coil in the center of the case, force the cooled air out on the floor, and then pull it vertically into a flue at the top of the case. The main point to keep in mind is that a high humidity must be maintained and the intensity of the air cannot be directed on the products.

Single installations of one condensing unit and one case usually give best results. The refrigerant is fed to the coils through a metering device, such as a thermostatic expansion valve or capillary tube similar to that used on domestic refrigerators. The control may be either of the pressure or temperature type. The recent development of a control that cuts in by coil temperature and cuts out by case temperature is being used successfully.

Temperatures generally accepted for meat display cases are 40 to 45 F on the mezzanine shelf, 36 to 38 F on the main shelf, and 34 to 36 F in the base. Frequent cycling with small variation in temperature and humidity between on-and-off cycles is desirable for best results. A long on-cycle tends to pull the moisture out of the meat and a long off-cycle tends to put it back on the surface of the meat. The first results in dehydration and the second in sliming.

The heat load on a refrigerated display case will vary with its construction. Those cases well insulated and with the sash properly sealed will have less heat loss. The accepted method of determining heat loads is to figure the actual heat leakage through the walls and through the glass, and add an allowance of from 25 to 60% to take care of service. Thirty-five per cent added for a service load is average. Another method to determine the heat leak is to place a bank of lamps inside the case in a controlled ambient with an integrating watt meter connected with the lines to the lamps. The amount of power required to maintain a definite temperature difference between inside and outside temperature will determine the heat leakage through the walls.

Delicatessen Cases

17. Delicatessen cases are distinguished from meat cases in that a varied type of product is kept in them requiring additional shelf space. For this reason, delicatessen cases are usually 54 in. high and are equipped with two mezzanine shelves in addition to the main shelf. They also have a storage compartment for storing meats, butter, milk, cheese, and bottled goods. Standard lengths are 6, 8 and 10 ft.

General construction details are similar to those of meat display cases; however, due to the added height the contour is different. The front glasses are wider and the sliding sash on the rear are more nearly vertical, allowing easy access to all the shelves. An average temperature of 40 to 42 F in the display compartment and a temperature of 38 to 40 F in the storage compartment is required.

Closed-Type Dairy Cases

18. Increased attention paid to the sale of dairy products has brought about a new design called dairy cases. The popular type is a self-serve upright model with large sliding display sash in the upper compartment and smaller sliding or reach-in doors in the storage compartment. Some manufacturers arrange the sash to raise vertically through the use of counter balances.

Dairy cases require good lighting and frequently an advertising sign is incorporated into the lighting fixture.

Temperatures of 36 to 45 F are satisfactory, depending on the product displayed. The coiling may be gravity type or forced air, and the machine should be selected to operate 16 hr in a 90 F room with service on the fixture.

One type being used in both large and small stores is shown in **Fig. 2.** This case is a semiportable, self-contained type, which can be moved from place to place in the store with little difficulty.

Closed-Type Vegetable Cases

19. Closed vegetable display cases are usually of the self-service type with sliding sash giving accessibility to the display compartment. They are built in 6, 8, 10, and 12-ft lengths. Most vegetable cases have storage compartments, equipped with solid sliding or swinging doors, underneath the display compartment.

One arrangement is similar to the upright self-service dairy case with the display sash vertical or almost vertical. Another popular arrangement is a low case with the main vegetable display shelves about 36 in. off the floor so that the customer has to reach forward only slightly to examine the contents of the shelf conveniently. The sliding sash on this case are almost in a horizontal position.

The construction of the vegetable case is similar to that of meat cases. If a porcelain liner is used, it should be acid-proof. Sliding sash may have hard rubber frames or a small metal binding edge around the glass, depending on the application.

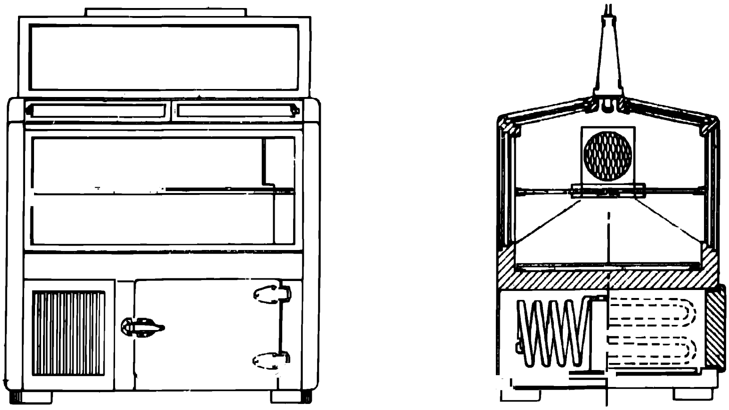


Fig. 2. Two Views of Portable Dairy Display Case

High levels of illumination are desirable to the proper display of vegetables in order to bring out the vivid colors. For this purpose fluorescent lights have proved very satisfactory.

The coiling may be gravity type or forced air. A very high relative humidity, and an average temperature of 38 to 45 F in the display compartment and 36 to 40 F in the storage compartment, are required. To accomplish this the machine is usually specified to operate not more than 16 hr per day in a 100 F ambient.

Closed-Type Candy Case

20. The closed-type candy case is a small self-contained type case 5 to 6 ft long, 30 in. deep and 36 in. high. It is designed to set against the wall or out in the store. The refrigerated storage compartment is accessible from a front reach-in door. It has a small, well-lighted display compartment for displaying candy. The coiling is usually forced-air type designed to maintain temperatures of 65 to 67 F and a relative humidity of approximately 70%. In view of the higher temperatures, insulation of 1 to 2 in. is considered satisfactory. The condensing units are usually the small hermetic type.

Closed-Type Frozen-Food Case

21. With the increasing popularity of frozen foods, display cases are a "must" in food stores. There are two types of closed frozen-food display cases—the type which displays the frozen foods but is not accessible to the customer, and the self-serve type which provides display and at the same time allows the customer to help himself. Both types are similar in construction to the frozen food chest.

Cases with an inclined display glass should have at least four thicknesses of glass to prevent fogging over. They should be well lighted and the coiling should be designed to keep the products displayed at the proper temperature.

The self-serve type case may have flat or inclined sliding doors, friction type or on rollers. The doors should be designed to prevent freezing and sticking. Good lighting is essential to good display and most manufacturers incorporate advertising and price display merit into the lighting fixture.

The closed-type frozen-food case with inclined glass display usually has a plate coil shelf for displaying the merchandise, and the coiling for the storage compartment as well as for the self-service case is similar to that of frozen-food chests. A temperature of 0 F is required for the safe storage of frozen foods.

The condensing unit should be specified to operate not more than 16 hr per day in a 90 F room.

Open-Type Self-Serve Display Cases

Open-type display cases are essentially wells of cold air into which the products are placed.

Open Vegetable Cases

22. The open type of self-service vegetable and fruit display and storage case is a development of the last decade. It is built both single-duty and double-duty in varying lengths, 6 to 12 ft long, and is approximately 40 in. high and 36 in. deep from front to back. It consists of little more than an open-top compartment or tank with insulated ends and back wall, and with glass front for display. Mirrors are placed above the back wall in a sloping position to multiply the display and give the impression of a greater mass of produce. Coiling may be plates placed around the inside lining, including the floor, or as dividers. Some manufacturers use a finned or plate coil on the back wall in conjunction with the liner coils to create a very slow and gentle circulation in the cabinet. Another arrangement is to install a fin coil in a vertical duct back of the mirrored section so that a thermal syphon system is created. The coils are so arranged as to maintain a line of refrigeration even with the top of the display.

The vegetables are displayed at such a level as to be most convenient for inspection by the customer, and at the same time to be seen for a long distance by any prospective customer.

With this type of case it is necessary to maintain a very high relative humidity to keep the product in its original condition and avoid dehydration.

Vegetables inside the refrigerated zone become chilled and the top, being entirely open to the warm air of the room, allows moisture in the warm air to be condensed on the vegetables. Contrary to most expectations, there is no great exchange between the warm and the cold air.

Since there is a gradient of temperature between the bottom and the top of the display section in the open type of case, due to the very little amount of circulation allowed, vegetables requiring lower temperatures should be stored in the lower part of the compartment, and those requiring the higher temperatures placed higher in the case. In general it can be said that the temperature range will vary from 35 to 46 F. A short running time and comparatively long off period on the operation cycle of the compressor are essential for proper refrigeration conditions.

High levels of illumination are desirable for the proper display of vegetables in order to bring out vivid colors. For this purpose fluorescent lights are popular and have proved very satisfactory. The same light for illuminating the display can be used to light an advertising display design.

Open Meat Cases

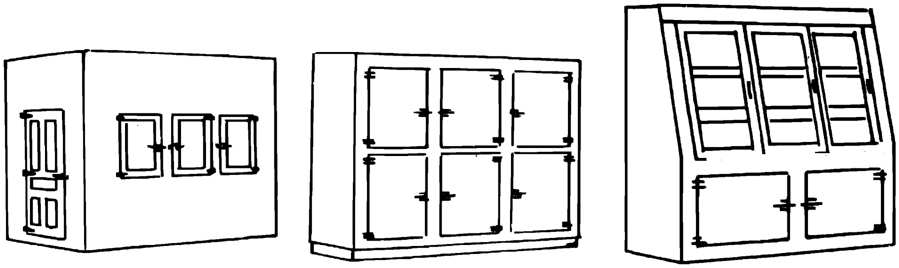
23. Most meat cases are 10 and 12 ft in length and are equipped with mirror top sections to enhance the display. The refrigeration may be accomplished with a coil located in a vertical duct behind an insulated mirror section, making a thermal syphon system, with plate coils in the bottom and around the sides, or with a combination of finned and plate coils. The temperature gradient from the bottom shelf to the top edge of the display glass will depend upon the type of coiling and the depth of the well. The top six inches is usually the buffer zone between the storage temperature and the room temperature and should be used only for short time display.

Since meats will be wrapped in moisture-proof paper a high humidity is not required. Lighting should be of the fluorescent type and of high intensity. This type of case should not be installed where fans or drafts will blow over the case, forcing the cold air out and the warm air in.

The condensing unit should be of sufficient size to maintain proper temperatures and at the same time cycle sufficiently to allow defrosting even under maximum conditions.

Open Dairy Case

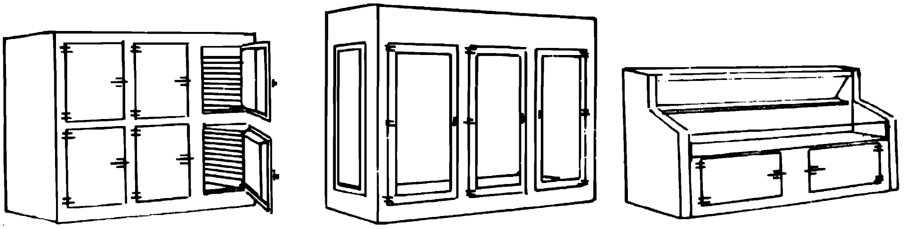
24. The open-type dairy case is similar to the meat case. In many cases it is the same with special step-type shelves to al-



Walk-in Cooler

Restaurant Refrigerator

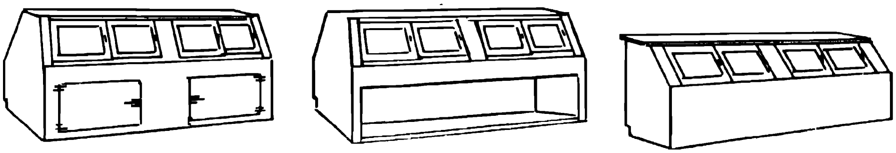
Dairy Refrigerator—Sliding Sash



Bakery Refrigerator

Florist Refrigerator

Fruit and Vegetable, Dairy, and Meat Case—Open Self-Serve Type



Meat Case—Double-Duty

Meat Case—Single-Duty

Chill Bar

Fig. 3. Various Types of Commercial Refrigerators

low building the display up higher. Dairy products such as milk and cream should be stored in the lower temperature region of the display compartment, whereas cheese, butter, etc., may be stored in the higher temperature area.

Later style dairy cases have two or even three refrigerated shelves.

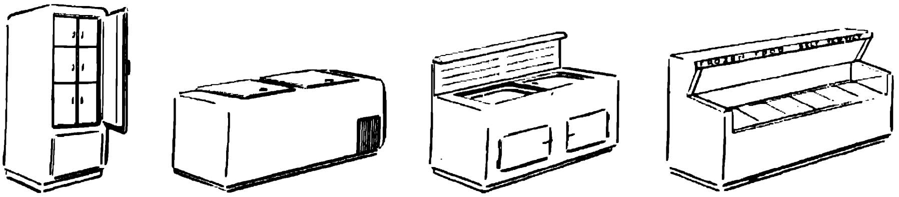
Open Candy Cases

25. Open-type self-service candy cases are a recent development. They are very similar in construction to the closed type with the addition of an open-type display above the storage compartment. Refrigeration in the display compartment is accomplished with plate coils on the floor and along the sides or with a fin coil at the back. Care must be taken to prevent condensation of moisture on the packages and also to protect the bottom of the packages from condensation or defrost water.

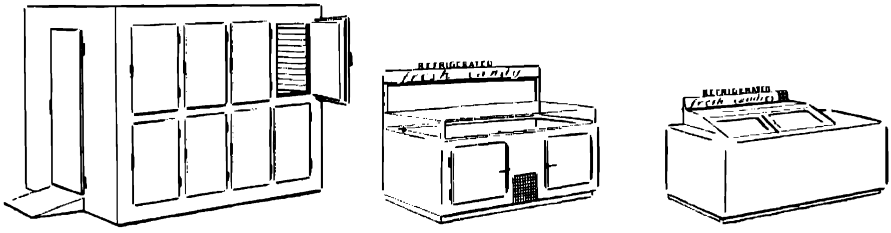
Open-Type Frozen-Food Display Cases

26. The advance made by the frozen-food industry during the past few years has made a definite demand for frozen-food cases. Freezer chests and ice-cream cabinets have been used for storing frozen foods, but they do not provide display or accessibility.

The design of an open-type frozen-food display case is similar to that of an open-type meat case. Since frozen foods require temperatures of −5 F to +5 F, the construction of these cabinets is necessarily



Frozen Food Refrigerator **Frozen Food Chest** **Closed Type Frozen Food Case** **Open Type Frozen Food Case**



Bakery Cooler **Open Type Candy Case** **Closed Type Candy Case**

Fig. 4. Types of Refrigerated Cabinets

much heavier. The insulation should be from 4 to 6 in. thick and must be sealed in so it is moisture tight. The exterior and interior lining around the top must be properly separated with suitable breaker strips to prevent heat flow.

The coiling should be designed to maintain proper temperatures and at the same time provide a buffer temperature between the outside air and the top of the display so that there will not be an excessive amount of condensation on the plates. Ice-cream freezer plate type coils, some equipped with eutectic hold-over solutions, are used. These coils may be made up of tubing which is soldered or mechanically fastened to plates, or they may be of the all-steel welded-plate type. Some manufacturers use a fin coil in conjunction with the plate coils to cool and dehydrate the air before it comes in contact with the plates.

The frozen food stored in the case should be stored low enough in the case to avoid being affected by the higher temperatures in the buffer area and the room temperature above. Some cases are provided with a lid that may be closed over night and over weekends. Several manufacturers are experimenting with an automatic defrosting arrangement that periodically defrosts the coils.

Since the product is already frozen when placed in the case it is not necessary to figure a product load. Only the heat leak through the walls and service are considered when selecting the condensing unit. It is most important to protect the case from drafts and fans. High intensity fluorescent lighting is used to light the display brilliantly; it may also be enhanced through the use of a sloping mirror section at the back or with vividly colored lithographs showing pictures of the various frozen foods.

34. SERVICING

ONCE the source of trouble in a refrigeration system is located, it is usually an easy matter to correct it, the main difficulty in servicing work being the location of the trouble. It is important that a **systematic** method of **analysis** be followed. It would be considerably better for the service man to take the first half hour to observe the arrangement and operation of the machine than to take a blind shot at the cause of the trouble and make any adjustment in the hope that it will correct the trouble. Such a procedure often adds trouble because the adjustment was not necessary or proper.

When the service man is following a systematic method of trouble shooting, he will first listen carefully to the **complaint** of the owner or operator of the equipment. This will give him his first clue as to the cause of the trouble. He should then carefully check the complaint to see if it is justified.

In some instances the cause of the trouble will be immediately apparent but in the large majority of cases some measuring of pressures, temperatures and running time will have to be made. In this latter case, it will be necessary to determine the type of machine, since the method of locating the trouble will depend upon the type of the refrigerating machine and the method of control.

Classification of Complaints

For the purpose of this discussion, refrigerating machines have been classified both by the method of refrigerant control and motor control. The refrigerant **control methods** include thermostatic expansion valve, automatic expansion valve, low-side float, high-side float and capillary tube. Motor controls are classified as to temperature and pressure.

Owner's or operator's complaints may also be classified. These complaints are, in the order they are taken up; (1) Refrigerated space too warm, (2) refrigerated space too cold, (3) refrigerating machine runs too long, (4) water bill too high, (5) electric bill too high, (6) refrigerating system too noisy and (7) odors in refrigerated space.

Before a man does any trouble shooting he should be familiar with the operation of the various systems under all kinds of conditions. The basic systems of controls as shown in **Table 1** are especially important. He should know how a system normally operates and what pressures, temperatures and running times to expect under such conditions so that he will readily recognize the abnormal.

To determine the source of the trouble it is desirable to make observations as follows:

1. The temperature of the refrigerated space or of the substance cooled
2. Temperature of the cooling unit
3. Suction pressure
4. Head pressure
5. Temperature of liquid and suction line
6. Sound of refrigerant control valve
7. Running time
8. Noise.

Temperature of Refrigerated Space

The temperature of the cooling unit cannot be determined exactly, but an approximation can be secured by clamping an accurate thermometer to the surface. It will be found that the temperature of the surface will be about 10 deg F higher than the temperature of the refrigerant in the evaporator, as shown by the suction pressure when the condensing unit is running.

The suction pressure will give an indication of the temperature of the refrigerant in the cooling coil. If the suction pressure reading is made at the condensing unit and the suction lines are long, the pressure in the cooling unit will be from 1 to 5 psi lower than at the evaporator.

Table 1. Types of Refrigerant Controls

Refrigerant control	Application	Type evaporator	Condensing unit control	Features
Capillary tube	Domestic only	Flooded	Temperature	No moving parts; self-compensating; must have exact refrigerant charge.
Low-side float	Commercial Domestic Multiple systems	Flooded	Pressure-temperature	Simplicity; good heat transfer; evaporator must be kept level; requires comparatively large amount of space and refrigerant.
High-side float	Domestic Commercial	Flooded Dry	Pressure-temperature	Wear on needle and seat is slight because liquid is handled most of time; must have exact refrigerant charge.
Automatic expansion	Domestic Commercial	Dry	Temperature	Requires small space. Easily adjusted. Permits use of easily constructed cooling coil; must be used with steady loads.
Thermostatic expansion	Domestic Commercial Multiple systems	Flooded Dry	Pressure-temperature	Permits use of easily constructed cooling coil; utilizes full length of coil; requires comparatively small amount of refrigerant.

Normally it will be found that in retail fixtures the temperature differential between the refrigerant and air will be from 17 to 27 F. For wholesale storage of eggs, poultry, cheese and the like, the temperature differential should be for normal operation 14 to 17 F; for cut meats, vegetables, fruits, beer in wooden kegs 18 to 22 F; for meats in carcasses and some fruits 21 to 27 F; and for goods where humidity control is not important, such as bottled beverages, food in cans and beer in steel kegs, 27 to 37 F.

Head Pressure

In normal operation it will be found that the temperature of the condensing refrigerant will be about 30 to 35 F higher than the temperature of the air entering the condenser in an air-cooled condensing unit and about 20 to 25 F higher than the water entering a water-cooled unit when the units are operating. These figures are for suction temperatures of from 10 to 20 F. Lower suction temperatures will result in lower head pressures and temperatures while higher suctions will result in higher head pressures and temperatures.

Fig. 1 shows the approximate head pressure of air-cooled condensing units using Freon-12, methyl chloride and sulfur dioxide.

Liquid and Suction Line Temperatures

The temperature of the **liquid line** will give an indication of the condition of the refrigerant within the liquid line. Normally the temperature of the liquid line is slightly above the temperature of the air or water leaving the condenser. If it is noted that the line is considerably warmer, this condition is probably due to the fact that the line is filled with gas rather than liquid due either to improper operation of the condenser or a shortage of refrigerant in the system. If, on the other hand, the line is much cooler than the leaving temperature of the condenser air or water, expansion of the refrigerant is probably taking place in the liquid line. This condition is caused by too great a pressure drop due to too long a line, too small a tube, excessive lift or a restriction such as a kink or partially closed valve.

The **suction line** temperature is normally a trifle lower than the room temperature but it will fall somewhat as the amount of refrigerant circulating is increased. If the line is considerably colder than normal or is frosted, it is an indication that liquid refrigerant is entering the suction line. Such a condition could be caused by improper operation of the refrigerant control **valve.**

The temperature of the **compressor** will give a good indication of the operation of the system. Normally the compressor head is fairly hot to the touch unless it is water-cooled. If it is not water-cooled and if it is fairly cool there is a good possibility that for some reason or other not enough refrigerant is being circulated. If it is very hot it will be found that the head pressure is excessive due to poor condensation or to air in the system.

The base of the compressor normally will be slightly above room temperature. A very warm compressor base may indicate a **shortage of oil** while a cool base may be caused by excessive oil or liquid refrigerant being slugged back through the suction line from the evaporator.

Normally the refrigerant **control valve** is fairly quiet in operation, there being only a slight noise due to the refrigerant passing through the orifice. However, when gas is passing through the valve instead of liquid a noticeable hissing noise will be heard. If the valve is completely quiet it is a good indication that no refrigerant is passing through.

Automatic condensing units will run from 5 to 20 hr or more a day, depending on a number of factors such as air temperature, design of unit and system and the setting of controls. Extremely short cycles or very long periods of continuous operation should be noted as they are indications of improper operation.

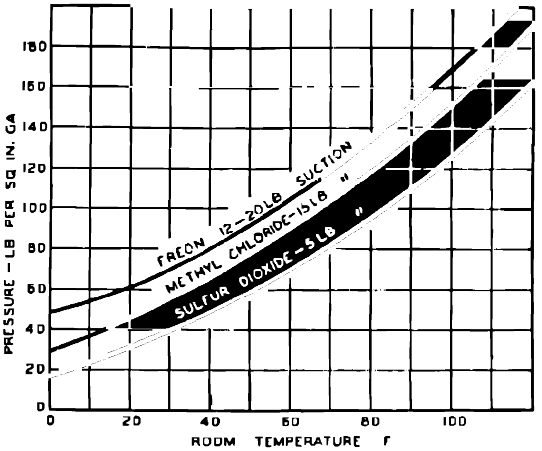


Fig. 1. Approximate Head Pressure of Air-Cooled Condensing Units Using Freon-12, Methyl Chloride and Sulfur Dioxide

Unusual noises in the compressor, motor, drive or refrigerant control valve should be noted as they serve as a good indication of trouble.

Trouble Shooting Charts

The following trouble shooting charts are not intended to serve as an inflexible guide for the service man, but they are designed merely to show one method of approach. Nothing can take the place of a fundamental knowledge of refrigeration and of the normal operation of refrigeration equipment in trouble shooting work. Yet these charts should be of considerable help not only to the man just starting in servicing but also to the more experienced man in pointing out how a logical method of approach will simplify trouble shooting.

The first observation to be made by the service man is how the condensing unit is operating, for example, whether it is operating at all, short cycling or operating continuously. The next observation is made to determine the type of system, that is, whether it is a high-side or low-side float or thermostatic or automatic expansion valve system and whether it is pressure or temperature controlled.

Once the method of operation and type of system are determined it is then necessary to make detailed observations on the condition of the system. These include the suction pressure, head pressure, temperature of the evaporator, compressor, motor, suction and liquid line, and sound of refrigerant control valve.

It will be noted that the charts are so arranged that all the troubles which apply to any type of system are first taken up, then those applying only to systems with pressure controls, then ones applying to systems with thermostatic or temperature controls, and finally those applying only to particular systems. This arrangement not only considerably reduces the size of the charts but also makes them easier to use.

COMPLAINT: REFRIGERATED SPACE TOO WARM—I

First Observation: Condensing Unit Not Operating
Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—high superheat	No current to condensing unit due to:	
	1. Open switch	1. Close switch
	2. Fuse blown	2. Replace fuse
	3. Power lines dead	3. Have lines repaired
	4. Broken connection	4. Repair
	Suction, temperature, or head pressure control	
	1. Control out of adjustment	1. Adjust
	2. Control contacts corroded	2. Clean with sandpaper
	3. Broken mercury bulb	3. Replace bulb or control
	4. Spring broken or worn out —control stuck open	4. Replace spring or control
	5. Control out-of-level	5. Level
	6. Leaky bellows or joints	6. Repair or replace
	Motor overload switch open due to:	
	1. Belt driving compressor is too tight	1. Loosen belt
	2. Failure of holding mechanism or dirty contact points	2. Replace or repair
	3. Low line voltage	3. Decrease load on line or increase wire size
	4. Bearings too tight or lack of oil	4. Oil or repair
	5. Compressor stuck	5. Repair or replace
	Motor failure	Repair or replace

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—high Evaporator—high superheat	High head pressure cut-out or motor overload switch open due to high head pressures caused by: 1. High air temperature entering air cooled condenser 2. Condensing water temperature too high 3. Air in system 4. Clogged condenser	1. Use an evaporative condenser or cooler supply of air 2. Use an evaporative condenser 3. Purge system 4. Clean or replace condenser
Suction pressure—below normal cut-in point Head pressure—low Evaporator—high superheat	Space surrounding condensing unit is too cold	Partially enclose condensing unit in box. Block off part of condenser

Type System: Any with Pressure Control

Suction pressure—below normal cut-in point Head pressure—below normal Evaporator—high superheat	An obstruction preventing refrigerant from circulating 1. Closed liquid line service valve 2. Liquid line stopped by kink 3. Strainer or dehydrator plugged 4. Liquid line solenoid valve failure 5. Suction service valve closed 6. Suction line stopped by kink in line or at flare nut 7. Ice, wax or dirt plugging refrigerant metering valve 8. Refrigerant charge lost	1. Open 2. Remove kink 3. Clean 4. Repair or replace 5. Open 6. Remove kink 7. Clean valve. Install dehydrator. Replace oil with oil of lower wax content 8. Repair leak and recharge

Type System: Thermostatic Expansion Valve—Pressure Controlled

Suction pressure—below normal cut-in point Head pressure—below normal Evaporator—high superheat	1. Power element has lost its charge 2. Expansion valve out of adjustment	1. Replace expansion valve 2. Adjust

Type System: High-Side Float—Pressure Controlled

Suction pressure—below normal cut-in point Head pressure—below normal Evaporator—high superheat	1. Float has leak in it and sinks in refrigerant	1. Replace

COMPLAINT: REFRIGERATED SPACE TOO WARM—II

First Observation: Condensing Unit Short Cycles

Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—high superheat Also:	If motor overload switch is tripping	
1. Motor hot	1. Belt driving compressor too tight	1. Loosen belt
2. Lights dim when motor starts	2. Low line voltage	2. Decrease load on line or use larger wire
3. Motor hot	3. Motor too small	3. Replace
4. Motor hot	4. Bearings lack oil	4. Oil
5. Motor may be hot	5. Ground or short circuit in motor winding	5. Repair or replace
Suction pressure—above normal cut-in point Head pressure—high Evaporator—high superheat	Head pressure cut-out or motor overload switch opening due to high head pressure caused by:	
1. Receiver pressure low. Motor and compressor head hot. Liquid line warmer than normal	1. Partially closed discharge service valve	1. Open valve
2. Motor and compressor head hot. Liquid line warmer than normal	2. Air-cooled condenser	2.
	(a) Dirty	(a) Clean
	(b) Air temperature too high	(b) Install evaporative or water-cooled condenser
	(c) Poor air circulation	(c) Remove obstruction
3. Motor and compressor head hot. Liquid line warmer than normal	3. Water-cooled condenser	3.
(a) No water discharged to sewers	(a) No water	(a) Remove obstruction in water pipe
(b) Discharge water very warm	(b) Restricted water flow	(b) Increase pressure or pipe size, remove obstruction
(c) Discharge water very warm and quantity excessive	(c) Water temperature too high	(c) Install evaporative condenser
(d) Discharge water cool and quantity excessive	(d) Corroded condenser	(d) Clean or replace
4. Motor and compressor head hot. Liquid line warmer than normal	4. Evaporative cooled condenser	4.
(a) Hot dry air leaving condenser	(a) No water	(a) Correct trouble in water line or float valve
(b) Hot dry air leaving condenser	(b) Spray nozzle clogged	(b) Clean
(c) Hot dry air leaving condenser	(c) Pump not operating	(c) Repair
(d) Same as (c)	(d) Coil surface dirty	(d) Clean

Observations	Trouble	Remedy
(e) No air leaving condenser	(e) Air inlet or outlet obstructed	(e) Remove obstruction
(f) No air leaving condenser	(f) Fan not operating	(f) Repair
5. Motor and compressor head hot. Liquid line warmer than normal	5. Air in system	5. Purge system
6. Motor and compressor head hot. Liquid line warmer than normal and compressor may be jumpy and slug at start	6. Too much refrigerant	6. Release some refrigerant
7. Receiver pressure low. Motor and compressor head hot. Liquid line warmer than normal	7. Restricted condenser tube	7. Clean or replace
8. Motor and compressor head hot. Liquid line warmer than normal	8. Condenser too small	8. Replace
Suction pressure—normal Evaporator—normal superheat Head pressure—low	1. Evaporator too small 2. Evaporator surface dirty	1. Install larger evaporator 2. Clean
Suction pressure—rises rapidly when compressor stops Evaporator—suction line frosted Head pressure—low	Compressor is removing refrigerant faster than it can be evaporated due to restricted air flow to evaporator or ice on evaporator.	Remove obstruction to air flow. Defrost

COMPLAINT: REFRIGERATED SPACE TOO WARM—III

First Observation: Condensing Unit Short Cycles

Type System: Any with Pressure Control

Observations	Trouble	Remedy
Suction pressure—rises rapidly when compressor stops Head pressure—low Evaporator—high superheat Also:	An obstruction in liquid or suction line is reducing flow of refrigerant or there is a refrigerant shortage	Remove obstruction. Add refrigerant
1. Hissing at metering valve	1. Shortage of refrigerant	1. Add refrigerant
2. Hissing at metering valve. Liquid line temperature decreases with height above receiver	2. Liquid line too small or too great a lift	2. Replace with larger line. Subcool liquid or install heat interchanger
3. Hissing at metering valve and temperature drop across obstruction	3. Strainer or dehydrator partially plugged or too small	3. Clean strainer. Replace dehydrator charge. Replace strainer or dehydrator
4. Hissing at metering valve and temperature drop across obstruction	4. Liquid line partially closed by kink or other obstruction	4. Remove kink or obstruction
5. Hissing at metering valve. Cold liquid line	5. Liquid service valve partially closed or too small	5. Open or replace

REFRIGERATED SPACE TOO WARM III (Continued)

Observations	Trouble	Remedy
6. Suction pressure in evaporator is much higher than at compressor	6. Suction line partially stopped by kinks or other obstruction, or too small	6. Remove obstruction. Replace with larger line
7 Hissing at metering valve and temperature drop across solenoid valve	7. Liquid line solenoid valve too small	7. Replace
	8. Ice, wax or dirt partially closing metering valve	8. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
	9. Differential on control too small	9. Increase differential

Type System: Any with Temperature Control

Suction pressure—rises rapidly when compressor stops Head pressure—low or normal Evaporator—high superheat	1. Differential on control too small	1. Increase differential

Type System: Thermostatic Expansion

Suction pressure—rises rapidly when compressor stops Head pressure—low or normal Evaporator—high superheat		
1. Test for ice—allow expansion valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing large amount of liquid to flow through in a short time	1. Ice, wax or dirt partially closing expansion valve	1. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
2. Same	2. Expansion valve orifice too small	2. Replace with larger orifice
3. Try a short turn of expansion valve adjusting nut first. If this is no good, something else is wrong	3. Expansion valve out of adjustment	3. Adjust
4. Warming feeler bulb raises suction pressure and decreases superheat only slightly	4. Expansion valve power element has partially lost charge	4. Replace expansion valve
5. Warming power element with hands corrects trouble temporarily	5. Expansion valve in too cold a location and power element is cooler than feeler bulb	5. Move expansion valve to warmer location

Type System: High-Side Float with Pressure Control

1. Evaporator partially sweated or frosted	1. Shortage of refrigerant	1. Add refrigerant

COMPLAINT: REFRIGERATED SPACE TOO WARM—IV

First Observation: Condensing Unit Operates Continuously
Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in pressure Head pressure—low Evaporator—high superheat	Refrigerant is not circulating at proper rate due to obstruction, inefficient compressor or shortage of refrigerant	
1. Compressor head and liquid line cooler than normal	1. Compressor belt slipping	1. Tighten belt
2. Hissing at refrigerant metering valve. Liquid line cooler than normal	2. Liquid line service valve partially closed or too small	2. Open or replace
3. Hissing at refrigerant metering valve. Temperature drop across strainer or dehydrator	3. Strainer or dehydrator partially plugged or too small	3. Clean strainer. Install new dehydrator charge Replace
4. Hissing at refrigerant metering valve. Temperature drop across solenoid valve	4. Liquid line solenoid valve too small	4. Replace
5. Hissing at refrigerant metering valve. Temperature drop across obstruction	5. Liquid line partially closed by kink or other obstruction	5. Remove obstruction
6. Hissing at refrigerant metering valve and gradual drop in temperature of liquid line	6. Liquid line too small or too great a lift	6. Replace with larger line. Subcool liquid or install heat interchanger
7. Suction pressure in evaporator is much higher than at compressor	7. Suction line partially stopped by kinks or other obstruction, or too small	7. Remove obstruction. Replace with larger line
8. Hissing at refrigerant metering valve. Compressor head cooler than normal	8. Shortage of refrigerant	8. Add refrigerant
9. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt, flush valve by allowing a large amount of refrigerant to flow through valve in short time. Repeat a number of times	9. Ice, wax or dirt partially closing refrigerant metering valve	9. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
10. Liquid line and compressor head cooler than normal	10. Inefficient compressor (a) Leaky compressor valves (b) Leaky piston rings (c) Leaky gaskets between cylinders	10. (a) Repair (b) Repair (c) Repair
11. Compressor speed too low	11. Low line voltage	11. Decrease load on line or increase line size
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—high superheat	Condensing unit capacity too small	Install larger condensing unit

REFRIGERATED SPACE TOO WARM IV (Continued)

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—high Evaporator—high superheat	Condensing unit has reduced capacity due to high head pressure caused by:	
1. Receiver pressure low. Motor and compressor head hot	1. Partially closed discharge service valve	1. Open valve
2. Motor and compressor head hot. Liquid line warmer than normal	2. Air-cooled condenser	2.
	(a) Dirty	(a) Clean
	(b) Air temperature too high	(b) Install evaporative or water-cooled condenser
	(c) Poor air circulation	(c) Remove obstruction
3. Motor and compressor head hot. Liquid line warmer than normal	3. Water-cooled condenser	3.
(a) No water discharge to sewer	(a) No water	(a) Remove obstruction in water line
(b) Discharge water very warm	(b) Restricted water flow	(b) Increase pressure or pipe size, remove obstructions
(c) Discharge water very warm and quantity excessive	(c) Water temperature too high	(c) Install evaporative condenser
(d) Discharge water cool and quantity excessive	(d) Corroded condenser	(d) Clean or replace
4. Motor and compressor head hot. Liquid line warmer than normal	4. Evaporative cooled condenser	4.
(a) Hot dry air leaving condenser	(a) No water	(a) Correct trouble in water line or water valve
(b) Hot dry air leaving condenser	(b) Spray nozzle clogged	(b) Clean
(c) Hot dry air leaving condenser	(c) Pump not operating	(c) Repair
(d) Same as (c)	(d) Coil surface dirty	(d) Clean
(e) No air leaving condenser	(e) Air inlet or outlet obstructed	(e) Remove obstruction
(f) No air leaving condenser	(f) Fan not operating	(f) Repair
5. Motor and compressor head hot. Liquid line warmer than normal	5. Air in system	5. Purge system
6. Motor and compressor head hot. Liquid line warmer than normal and compressor may be jumpy and slug at start	6. Too much refrigerant	6. Release some refrigerant
7. Receiver pressure low. Motor and compressor head hot	7. Restricted condenser tube	7. Clean or replace
8. Motor and compressor head hot. Liquid line warmer than normal	8. Condenser too small	8. Replace

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—normal superheat	Condensing unit overloaded due to:	
	1. Hot foods	1. Instruct owner in proper operation
	2. Doors left open	2. Instruct owner in proper operation
	3. Loose fitting doors	3. Replace
	4. Poor gaskets	4. Replace
	5. Poor insulation	5. Replace
	6. Condensing unit and coil too small	6. Replace

Type System: Any—Temperature Controlled

Suction pressure—very low and does not rise when compressor is stopped Head pressure—low Evaporator—high superheat	An obstruction is preventing the flow of refrigerant	
1. Liquid line cooler than normal	1. Liquid line service valve closed	1. Open
2. Liquid line cooler than normal	2. Strainer or dehydrator completely plugged	2. Clean strainer. Install new dehydrator charge
3. Liquid line cooler than normal	3. Liquid line closed by kink or other obstruction	3. Remove kink or obstruction
4. Liquid line cooler than normal. Test for ice—allow valve to warm up to melt ice. Tap lightly to free ice or dirt	4. Ice, wax or dirt completely closing refrigerant metering valve	4. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
4. Liquid line cooler than normal	5. Refrigerant metering valve closed due to mechanical failure	5. Repair or replace
6. Suction pressure in evaporator is much higher than at compressor	6. Suction line completely stopped by kink, closed service valve or other obstruction	6. Remove kink or obstruction, open valve
Suction pressure—high Head pressure—normal or low Evaporator—warm Suction line—sweating Hissing at metering valve	Metering valve stuck open and so much refrigerant enters low side that it cannot evaporate at a pressure low enough to give a low temperature	Repair or replace

Type System: Thermostatic Expansion

Suction pressure—above normal cut-in point Head pressure—low Evaporator—high superheat	Refrigerant is not circulating at proper rate due to obstruction	

REFRIGERATED SYSTEM TOO WARM IV (Continued)

Observations	Trouble	Remedy
1. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing large amount of liquid to flow through valve in short time. Repeat a number of times	1. Ice, dirt or wax partially closing expansion valve	1. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
2. Try turning expansion valve adjusting nut slightly. If this does no good something else is wrong	2. Expansion valve out of adjustment	2. Adjust
3. Warming feeler bulb raises suction pressure and decreases superheat only slightly	3. Expansion valve power element has partially lost charge	3. Replace expansion valve
4. Warming power element with hands corrects trouble temporarily	4. Expansion valve in too cold a location. Power element is colder than feeler bulb	4. Move expansion valve to warmer location
Suction pressure—above normal cut-in point pressure Head pressure—low Evaporator—sweating or frosting suction line Crankcase—sweating	Expansion valve stuck open	Repair or replace

Type System: High-Side Float—Temperature Control

Suction pressure—very low and rises slightly when compressor is shut down Head pressure—low Evaporator—high superheat	Shortage of refrigerant	Add refrigerant
Suction pressure—very low and does not rise when compressor is shut down Head pressure—very high Evaporator—high superheat	High head pressure has caused float to collapse and sink, closing refrigerant valve	Determine cause of high head pressure and replace float

COMPLAINT: REFRIGERATED SPACE TOO COLD

First Observation: Condensing Unit Operating Too Long

Type System: Any

Observations	Trouble	Remedy
Suction pressure—below normal cut-in point Head pressure—normal Evaporator—too cold—normal superheat	1. Pressure or temperature control set too low 2. Pressure or temperature control defective—will not cut out	1. Adjust 2. Repair or adjust

Type System: Any—Pressure Controlled

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—no superheat; frosted or sweating suction line	1. Dirt, wax or ice on valve seat	1. Clean. Use oil of lower wax content. Install dehydrator

Type System: Thermostatic Expansion—Pressure Controlled

Suction pressure—above normal cut-in point Head pressure—normal Evaporator—no superheat; frosted or sweating suction line	1. Expansion valve out of adjustment	1. Adjust
	2. Expansion valve feeler bulb loosely clamped to suction line	2. Fasten securely

COMPLAINT: CONDENSING UNIT OPERATES TOO LONG

First Observation: Refrigerated Space Cold Enough

Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—low Evaporator—superheat slightly higher than normal Also:	Refrigerant is not circulated at proper rate due to an obstruction, inefficient compressor or shortage of refrigerant	
1. Compressor head and liquid line cooler than normal. Lights dim when motor starts	1. Low line voltage	1. Decrease load on line or install larger line
2. Compressor head and liquid line cooler than normal	2. Compressor belt slipping	2. Tighten belt
3. Hissing at refrigerant metering valve. Liquid line cooler than normal	3. Liquid line service valve partially closed or too small	3. Open or replace
4. Hissing at metering valve. Temperature drop across strainer or dehydrator	4. Strainer or dehydrator partially plugged or too small	4. Clean strainer. Install new dehydrator charge. Replace
5. Hissing at metering valve Temperature drop across solenoid valve	5. Liquid line solenoid valve too small	5. Replace
6. Hissing at metering valve. Temperature drop across obstruction	6. Liquid line partially closed by kink or other obstruction	6. Remove obstruction
7. Hissing at metering valve and gradual drop in temperature of liquid line	7. Liquid line too small or too great a lift	7. Replace with larger line. Subcool liquid or install heat interchanger
8. Suction pressure in evaporator is much higher than at compressor	8. Suction line partially stopped by kinks or other obstruction or too small	8. Remove kinks. Replace with larger line

CONDENSING UNIT OPERATES TOO LONG (Continued)

Observations	Trouble	Remedy
9. Hissing at metering valve. Compressor head cooler than normal	9. Slight shortage of refrigerant	9. Add refrigerant
10. Liquid line and compressor head cooler than normal	10. Inefficient compressor (a) Leaky compressor valve (b) Leaky piston rings (c) Leaky gaskets between cylinders	10. Repair or replace compressor
11. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing a large amount of refrigerant to flow through valve in short time. Repeat a number of times	11. Ice, wax, or dirt partially closing metering valve	11. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
12. Same	12. Compressor speed too low	12. Speed up by changing pulley size or using larger motor
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—superheat higher than normal	Condensing unit capacity too small	Install larger condensing unit
Suction pressure—above normal cut-in point Evaporator—superheat slightly higher than normal	Condensing unit has reduced capacity due to high head pressure caused by:	
1. Receiver pressure low. Motor and compressor head hot	1. Partially closed discharge service valve	1. Open valve
2. Motor and compressor head hot. Liquid line warmer than normal	2. Air-cooled condenser	2.
	(a) Dirty	(a) Clean
	(b) Air temperature too high	(b) Install evaporative or water-cooled condenser
	(c) Poor air circulation	(c) Remove obstruction
3 Motor and compressor head hot. Liquid line warmer than normal	3. Water-cooled condenser	3.
(a) No water discharged to sewer	(a) No water	(a) Remove obstruction in water pipe
(b) Discharge water very warm and quantity excessive	(b) Restricted water flow	(b) Increase pressure or pipe size, remove obstruction
(c) Discharge water very warm and quantity excessive	(c) Water temperature too high	(c) Install evaporative condenser
(d) Discharge water cool and quantity excessive	(d) Corroded condenser	(d) Clean or replace

Observations	Trouble	Remedy
4. Motor and compressor head hot. Liquid line warmer than normal	4. Evaporative cooled condenser	4.
(a) Hot dry air leaving condenser	(a) No water	(a) Correct trouble in water line or float valve
(b) Hot dry air leaving condenser	(b) Spray nozzle clogged	(b) Clean
(c) Hot dry air leaving condenser	(c) Pump not operating	(c) Repair
(d) Same as (c)	(d) Coil surface dirty	(d) Clean
(e) No air leaving condenser	(e) Air inlet or outlet obstructed	(e) Remove obstruction
(f) No air leaving condenser	(f) Fan not operating	(f) Repair
5. Motor and compressor head hot. Liquid line warmer than normal	5. Air in system	5. Purge system
6. Motor and compressor head hot. Liquid line warmer than normal and compressor may be jumpy and slug at start	6. Too much refrigerant	6. Release some refrigerant
7. Receiver pressure low. Motor and compressor head hot	7. Restricted condenser tube	7. Clean or replace
8. Motor and compressor head hot. Liquid line warmer than normal	8. Condenser too small	8. Replace
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—normal superheat	Condensing unit overloaded due to:	
	1. Hot foods	1. Instruct owner in proper operation
	2. Doors left open	2. Instruct owner in proper operation
	3. Loose fitting doors	3. Replace
	4. Poor gaskets	4. Replace
	5. Poor insulation	5. Replace
	6. Condensing unit and coil too small	6. Replace

Type System: Thermostatic Expansion

Suction pressure—above normal cut-in point Head pressure—low Evaporator—high superheat	Refrigerant is not circulating at proper rate due to obstruction	
1. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing large amount of liquid to flow through valve in short time. Repeat a number of times	1. Ice, dirt or wax partially closing expansion valve	1. Clean valve. Install dehydrator. Replace oil with oil of lower wax content

CONDENSING UNIT OPERATES TOO LONG (Continued)

Observations	Trouble	Remedy
2. Try turning expansion valve adjusting nut slightly. If this does no good something else is wrong	2. Expansion valve out of adjustment	2. Adjust
3. Warming feeler bulb raises suction pressure and decreases superheat only slightly	3. Expansion valve power element has partially lost charge	3. Replace expansion valve
4. Warming power element with hands corrects trouble temporarily	4. Expansion valve in too cold a location. Power element is colder than feeler bulb	4. Move expansion valve to warmer location

COMPLAINT: WATER BILL TOO HIGH

First Observation: Refrigerated Space Cold Enough

Type System: Any—Head Pressure Controlled Water Valve

Observations	Trouble	Remedy
Water flow—high Head pressure—normal	1. Water pressure too high	1. Install pressure regulating valve
	2. Water valve out of adjustment	2. Adjust
	3. Valve stuck open	3. Repair
	4. Condenser is dirty resulting in poor heat transfer	4. Clean or replace
Water flow—normal Head pressure—normal	1. Water valve leaks when shut off	1. Repair
	2. Condensing unit operates too long	2. See section entitled "Condensing Unit Operates Too Long"
Water flow—high Head pressure—high	Excessive water flow caused by high head pressure due to:	
	1. High water temperature	1. Install evaporative condenser
	2. Air in system	2. Purge
	3. Too much refrigerant	3. Remove some refrigerant
	4. Condenser is dirty resulting in poor heat transfer	4. Clean or replace
	5. Condenser too small	5. Replace
	6. Partially closed discharge or receiver valve	6. Open

Type System: Any—Electric Water Valve

Water flow—normal	1. Water valve leaks when shut off	1. Repair
	2. Condensing unit operates too long	2. See section entitled "Condensing Unit Operates Too Long"

Observations	Trouble	Remedy
Water flow—high	1. Water pressure too high	1. Install pressure regulating valve
	2. Water valve stuck open	2. Repair
	3. Water valve too large	3. Install smaller valve

COMPLAINT: ELECTRIC BILL TOO HIGH

First Observation: Refrigerated Space Cold Enough
Type System: Any

Observations	Trouble	Remedy
	1. Wrong electric rate	1. Get in touch with electric power company
	2. Machine operates too long	2. See section entitled "Condensing Unit Operates Too Long"

COMPLAINT: NOISE

Type System: Any

Observations	Trouble	Remedy
Compressor is noisy	1. Compressor foundation bolts loose	1. Tighten
	2. Drive belt is noisy	2.
	(a) Squeaks	(a) Clean. See if pulleys are in line. Use soap or belt dressing
	(b) Belt slaps because it is loose or worn	(b) Tighten
	3. Compressor bearings worn	3. Repair
	4. Valves noisy	4. Repair
	5. Squeaky seal	5.
	(a) Oil level too low	(a) Add oil and repair if necessary
	(b) Defective seal	(b) Replace or repair seal
	6. Compressor pumping oil	6. Remove excess oil
	7. Shipping blocks not removed	7. Remove
	8. Excessive refrigerant	8. Purge some refrigerant
	9. Air in system	9. Purge
	10. Compressor bearings worn	10. Repair
Motor is noisy	1. Motor loose on base	1. Repair or replace
	2. Bearings lack oil	2. Add oil
	3. Brushes squeak	3. Replace or fit brushes to commutator
	4. Motor bearings worn	4. Tighten bolts

COMPLAINT: ODORS

Type System: Any

Observations	Trouble	Remedy
Odor in refrigerated space	1. Cabinet dirty or spoiled food 2. Strong foods left uncovered 3. No trap in drain to sewer 4. Cabinet door kept closed when system is shut off 5. Refrigerant leaks 6. Brine leaks 7. Coil surface dirty	1. Clean and wash with strong solution of baking soda and water 2. Instruct owner to cover all strong foods 3. Install trap 4. Have owner leave door open when system is turned off 5. Locate and repair 6. Locate and repair 7. Wash with baking soda solution
Odors outside of cabinet	1. Refrigerant leaks 2. Brine leaks 3. Belt burned due to slipping 4. Motor overheated	1. Locate leak and repair 2. Locate leak and repair 3. Tighten belt. Replace if necessary 4. Locate cause of trouble Install overload relay

PART VI

AIR CONDITIONING

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35. AIR CONDITIONING

Part I. Theory of Air Conditioning

I. Properties of Mixtures of Dry Air and Steam ("Moist" Air)

1. The water vapor or steam mixed with dry air in the atmosphere is **superheated** when the relative humidity is less than 100% and saturated when the relative humidity is 100%. The **state** of a mixture of dry air and saturated steam is completely defined if the dry bulb temperature and total pressure of the mixture are known. The tables of Chapter 10 give the properties of mixtures of dry air and saturated steam for a total pressure of 29.921 in. Hg abs and for temperatures from −160 F to 200 F.

The state of a mixture of dry air and superheated steam is not completely defined by dry bulb temperature and total pressure. A third property of state is necessary, such as relative humidity, partial pressure of the water vapor, saturation ratio, dew point, or temperature of adiabatic saturation (also called thermodynamic wet bulb temperature).

2. Such a property in a mixture of dry air and water vapor is the **temperature of adiabatic saturation,** sometimes called the **thermodynamic wet bulb temperature.** If a mixture of dry air and water vapor flows over a large surface of water, the specific humidity of the mixture will be increased, and the mixture may be saturated. If the saturation process occurs at constant total pressure and with no external heat transfer, the temperature at which the mixture will be saturated depends upon the initial state of the mixture and upon the temperature at which the makeup water is supplied.

If the water is assumed to be supplied at the exit temperature of the saturated mixture, then this temperature, now defined as the temperature of adiabatic saturation, depends only upon the initial state of the mixture and is, therefore, a true property of state. The temperature of adiabatic saturation is also the temperature at which mixtures of air and water vapor may be saturated by contact with recirculated water in adiabatic processes of evaporative cooling.

The state of a mixture of dry air and water vapor is fixed and the various properties of state may be found when the total pressure of the mixture, the dry bulb temperature, and the temperature of adiabatic saturation are known. There is little difficulty in finding the first two of these quantities, but the **experimental determination** of the temperature of adiabatic saturation is more difficult. The temperature of adiabatic saturation of a mixture of air and water vapor is commonly taken as the temperature indicated by the **wet bulb thermometer** of a **sling psychrometer.** This procedure is quite accurate but entirely fortuitous, for the same procedure may involve large errors for other mixtures, particularly when the molecular weight of the gas is greatly different from that of the vapor.

To read accurately the temperature of adiabatic saturation of air-water vapor mixtures, the wet bulb thermometer must be swept by an air stream at a velocity of at least 500 fpm and must be exposed to radiation from surfaces that are at a temperature close to the dry bulb temperature of the air. A wet bulb thermometer in relatively stagnant air (air velocity of 20 fpm), when unshielded from radiation from surfaces at the dry bulb temperature, will read a temperature that is about 15% of the wet bulb depression **higher** than the temperature of adiabatic saturation; if shielded from radiation, the wet bulb thermometer in stagnant air will read a temperature that is about 15% **lower** than the temperature of adiabatic saturation.

For extremely high air velocities (5,000 fpm or more), the wet bulb thermometer, whether shielded or unshielded from surfaces at the dry bulb temperature, will read closely the temperature of adiabatic saturation.

The **energy balance** for the process of adiabatic saturation is

$$h+\frac{W'-W}{7{,}000}h_{f'}=h' \qquad (1)$$

where h = the enthalpy of the mixture of dry air and water vapor, Btu per lb of dry air
W = the specific humidity of the mixture, grains per lb of dry air
W' = the specific humidity for saturation at the temperature of adiabatic saturation, grains per lb of dry air
$h_{f'}$ = the specific enthalpy of the liquid (or solid) at the temperature of adiabatic saturation, Btu per lb
h' = the enthalpy of the saturated mixture at the temperature of adiabatic saturation, Btu per lb of dry air

The **saturation ratio** of the mixture, μ, also known as **percentage humidity** or **per cent saturation** is the ratio of the specific humidity of the mixture to the specific humidity for saturation at the dry bulb temperature, or

$$\mu=\frac{W}{W_s} \qquad (2)$$

Therefore,

$$h=h'-\frac{W'-\mu W_s}{7{,}000}h_{f'}$$

Also, the enthalpy of the mixture is given by the following equation

$$h=h_a+\mu(h_s-h_a)=h_a+\mu h_{as} \qquad (3)$$

where h_a = the enthalpy of the dry air in the mixture at the dry bulb temperature, Btu per lb of dry air
h_s = the enthalpy of the mixture of dry air and saturated steam at the dry-bulb temperature, Btu per lb of dry air

For simplicity in the use of Eq 3, it may be assumed with small error that the enthalpy of dry air in a mixture of dry air and superheated steam is equal to the enthalpy of dry air at the same total pressure and dry bulb temperature. Actually, because the partial pressure of the dry air in a mixture is less than the total pressure, the enthalpy of the dry air in the mixture slightly exceeds the enthalpy of dry air at a pressure equal to the total pressure.

It may be shown from the preceding equations that the saturation ratio is

$$\mu=\frac{h'-h_a-\dfrac{W'h_{f'}}{7{,}000}}{h_s-h_a-\dfrac{W_sh_{f'}}{7{,}000}} \qquad (4)$$

If the pressure-volume product of superheated steam is the same as the pressure-volume product of saturated steam at the same temperature, it may be shown that the volume of a mixture of dry air and superheated steam per pound of dry air is

$$v=v_a+\mu(v_s-v_a) \qquad (5)$$

where v_a = the specific volume of dry air at the total pressure and temperature of the mixture, cu ft per lb
v_s = the volume of the mixture of dry air and saturated steam at the dry bulb temperature, cu ft per lb of dry air

3. Examples will now be given to show how the properties of state of a mixture of dry air and superheated steam may be found from the tables of Chapter 10 covering properties of mixtures of dry air and saturated steam:

Find the saturation ratio, specific humidity, relative humidity, dew point, enthalpy and volume of a mixture of dry air and superheated steam when the dry bulb temperature is 80 F and the wet bulb temperature is 65 F if: (a) the total pressure is the standard atmospheric pressure of 29.921 in. Hg abs; (b) the total pressure is 29.00 in. Hg abs.

(a) From Table 1 (Chapter 10) at $t'=65$ F, find $W'=92.51$ grains per lb of dry air, $h_{f'}=33.1$ Btu per lb of steam and $h'=30.03$ Btu per lb of dry air; from the same table at $t=80$ F, find $h_a=19.23$ Btu per lb of dry air, $h_s=43.63$ Btu per lb of dry air, and $W_s=155.8$ grains per lb of dry air. The percentage humidity (saturation ratio) of this mixture may then be found from Eq 4 as follows:

$$\mu = \frac{30.03 - 19.23 - \dfrac{92.51(33.1)}{7,000}}{43.63 - 19.23 - \dfrac{155.8(33.1)}{7,000}} = 0.438$$

From Eq 2, the specific humidity of the mixture is then found to be

$$W = 0.438(155.8)$$
$$= 68.22 \text{ grains per lb of dry air}$$

The relative humidity, ϕ, is the ratio of the partial pressure of the steam in the mixture to the saturation pressure of steam at the dry-bulb temperature. It is permissible to interpolate in Table 1 (Chapter 10) on the basis of this specific humidity to find the partial pressure of the steam in this mixture to be 0.461 in. Hg. The relative humidity is

$$\phi = \frac{0.461}{1.0323} = 0.447$$

By a similar interpolation the dew point is found to be 56.6 F. This is the temperature at which condensation of the water vapor would begin if the mixture were cooled at the constant total pressure of 29.921 in. Hg.

The enthalpy of the mixture may be found from Eq 1 as

$$h = h' - \frac{W' - W}{7,000} h_{f'}$$
$$= 30.03 - \frac{24.29(33.1)}{7,000}$$
$$= 29.92 \text{ Btu per lb of dry air}$$

The volume of the mixture per lb of dry air is found from Eq 5 as

$$v = 13.602 + 0.438(14.09 - 13.602)$$
$$= 13.816 \text{ cu ft}$$

(b) When the total pressure of the mixture differs from the standard atmosphere, properties of the mixture of dry air and saturated steam at the given total pressure must be found first. At a total pressure of 29.0 in. Hg and a temperature of 80 F, properties of dry air are found as follows:

$$v_a = \frac{29.921}{29.0}(13.606) - 0.0042$$
$$= 14.034 \text{ cu ft per lb}$$
$$h_a = 19.327 - \frac{29.0}{29.921}(0.097)$$
$$= 19.233 \text{ Btu per lb}$$

Properties of a mixture of dry air and saturated steam at a pressure of 29.0 in. Hg and a temperature of 80 F follow:

$$v_s = \frac{13.606(29.921)}{29.0 - 1.0323} - 0.0042$$
$$= 14.552 \text{ cu ft per lb of dry air}$$
$$W_s = \frac{7,000\, v_s}{v_g} = \frac{7,000(14.552)}{633}$$
$$= 160.9 \text{ grains per lb of dry air}$$
$$h_s = 19.327 - \frac{(29.0 - 1.0323)0.097}{29.921}$$
$$+ \frac{160.9}{7,000}(1096.1)$$
$$= 44.43 \text{ Btu per lb of dry air}$$

In a similar manner, properties of a mixture of dry air and saturated steam at a pressure of 29.0 in. Hg and a temperature of 65 F may be shown to be $v_s = 13.942$ cu ft per lb of dry air, $W_s = 95.52$ grains per lb of dry air, and $h_s = 30.50$ Btu per lb of dry air.

The properties of the mixture of dry air and superheated steam follow:

$$\mu = \frac{30.50 - 19.233 - \dfrac{95.52(33.1)}{7,000}}{44.43 - 19.233 - \dfrac{160.9(33.1)}{7,000}} = 0.443$$

$$W = 0.443(160.9)$$
$$= 71.28 \text{ grains per lb of dry air}$$
$$v = 14.034 + 0.443(14.552 - 14.034)$$
$$= 14.333 \text{ cu ft per lb of dry air}$$
$$h = 30.50 - \frac{(95.52 - 71.28)}{7,000}(33.1)$$
$$= 30.39 \text{ Btu per lb of dry air}$$
$$\phi = 0.443\left(\frac{14.552}{14.333}\right) = 0.450$$

The partial pressure of the steam in the mixture is 0.450 (1.0323) or 0.4645 in. Hg, and the dew point, or saturation temperature of steam at this pressure, is 55.1 F.

II. Processes in Air Conditioning

4. a. Regardless of the nature of the process by which the state of a mixture of air and water vapor may be changed in conditioning the air, two important relations for such processes are the **conservation of weight** and the **conservation of**

energy equations. In drawing up a weight balance for steady flow conditions, the weight of all fluids entering the equipment (whether heater, cooler, etc.) must be equal to the weight of all fluids leaving in the same unit of time. In setting up the energy balance for steady flow of the fluids through the apparatus with no appreciable changes in elevation or velocity, the enthalpy of all fluids entering plus any heat or mechanical energy supplied (minus any heat or mechanical energy removed) may be equated to the enthalpy of all fluids leaving the apparatus.

b. The **fundamental processes** required in air conditioning are heating, cooling, humidifying, dehumidifying, and mixing; in addition, there are the processes of cleaning, supplying, exhausting, and distributing the air. The refrigerating engineer is usually concerned with maintaining the required dry bulb temperature and relative humidity within an enclosure where there is a gain of heat and a gain of water vapor. The required conditions may be maintained by direct expansion of a refrigerant in coils in the enclosure or by the supply of conditioned air. When the latter method is used, the conditioned air must be supplied at the proper rate and at a lower dry bulb temperature than the room air in order to balance the gain of heat; the specific humidity of the conditioned air must also be lower than that maintained in the enclosure, in order to balance the gain of water vapor.

Assume that a dry bulb temperature of t_r and a wet bulb temperature of t_r' must be maintained in an enclosure where there is a gain of heat of Q_h Btu per hr from transmission, solar effect, lights, people, processes, air change, etc., and a gain of water vapor of Q_v grains per hr from air change, processes, people, etc. If the conditioned air is supplied at the rate of M_a lb per hr (dry) and in a state a with a dry-bulb temperature of t_a and a specific humidity of W_a, the weight balance for water vapor is

$$M_a(W_r - W_a) = Q_v \qquad (6)$$

The energy balance is

$$M_a\left(0.24 + \frac{0.44W_a}{7,000}\right)(t_r - t_a) = Q_h \qquad (7)$$

From these relations, the required specific humidity of the conditioned air supplied to the enclosure is

$$W_a = W_r - \frac{(1,680 + 0.44W_r)(t_r - t_a)}{\dfrac{7,000Q_h}{Q_v} + 0.44(t_r - t_a)} \qquad (8)$$

For a given state of air in the enclosure and a given gain of heat and water vapor, fixing any one of the other three variables, (1) dry bulb temperature of conditioned air, (2) specific humidity of conditioned air, or (3) weight rate of supply of conditioned air, will immediately fix the remaining two.

Example. A room is to be maintained at a dry bulb temperature of 80 F and a wet bulb temperature of 67 F ($W_r = 78.3$). There is a gain of heat of 30,000 Btu per hr and a gain of water vapor of 105,000 grains per hr. If the dry bulb temperature of the conditioned air supplied to the room is 65 F, find the required specific humidity and rate of supply of conditioned air.

Solution. From (8) the required specific humidity is

$$W_a = 78.3 - \frac{(1714.5)(15)}{2,007}$$

$$= 65.5 \text{ gr per lb of dry air}$$

$$\text{From (6)} \quad M_a = \frac{105,000}{12.8}$$

$$= 8,200 \text{ lb of dry air per hr}$$

The state of the conditioned air that must be supplied to the room may be found on the psychrometric chart of Chapter 4 from the known dry bulb temperature (65 F) and the known specific humidity (65.5). The corresponding wet bulb temperature is 59 F; also, the volume of the supply air is 13.42 cu ft per lb of dry air, and the conditioned air must therefore be supplied at the rate of 1,830 cfm.

5. a. Some of the **processes** in air conditioning will be explained with reference to the psychrometric chart. **Adiabatic saturation or evaporative cooling** may be represented on the psychrometric chart without appreciable error by a process of constant wet bulb temperature. This process is exemplified by the humidification of air in spray-type equipment if the spray water is neither heated nor cooled. For example, if the initial dry bulb temperature of the air is 90 F, initial wet bulb 63 F, and if the dry bulb is reduced to 72.5 F by spraying the air with recirculated water, the initial and final properties of the mixture are:

	Initial	*Final*
Dry bulb temperature, F	90	72.5
Wet bulb temperature, F	63	63
Relative humidity, %	20	60
Dew point, F	44.5	57.6
Specific humidity, gr per lb	43	71
Enthalpy, Btu per lb	28.35	28.47

b. Cooling with dehumidification may be accomplished by passing air through a spray of water or over a surface maintained at a temperature below the initial dew point of the air. The problem is to select the surface and its temperature such that dehumidification and sensible cooling will be in the right proportion to balance the moisture and sensible heat gains in the conditioned enclosure. A skeleton psychrometric chart is shown in **Fig. 1** in which the specific humidity is plotted as ordinate vs. the dry bulb temperature as abscissa. State 1 represents the air introduced to the cooling surface, while t_s represents the temperature of that surface. If a point, S, is located on the saturation curve at the temperature t_s, the state of the air leaving the surface (one such point is represented by 2) will be on the straight line from 1 to S (with constant surface temperature).

The temperature at point 2 depends on the rate of flow of air, the area of the cooling surface, the sensible heat transfer coefficient of the surface, the surface temperature, and the dry bulb temperature of the entering air.

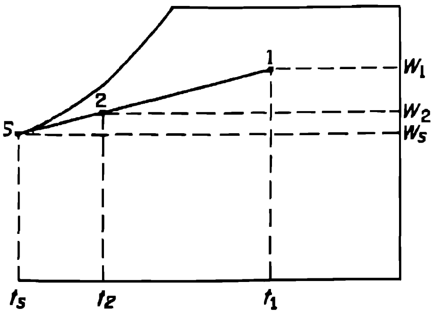

Fig. 1. Cooling and Dehumidification of Air

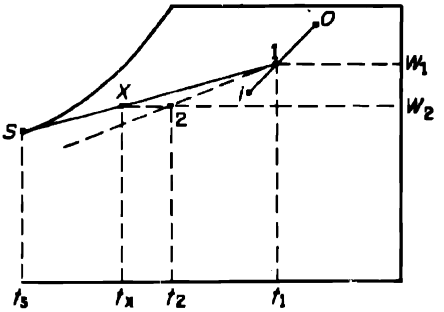

Fig. 2. Use of Reheat to Increase Dehumidification Ratio

The straight line joining the state of the air entering the air conditioner and that leaving it may fail to intersect the saturation curve as shown by the dashed line 1–2 in **Fig. 2**. This indicates that the ratio of latent to sensible heat loads is so great that at least two treatments must be used. With the state of the entering air fixed, the maximum ratio of latent to sensible cooling capacity obtainable with either spray- or surface-type equipment is fixed by the slope of the straight line drawn through this point tangent to the saturation curve. When the ratio of latent to sensible cooling loads is greater than can be handled in a single process, the air may be reheated after cooling. As shown in Fig. 2, the air may be cooled by a surface at t_s to some point X, where the desired specific humidity exists. In this cooling, there is a greater reduction in the dry bulb temperature than is desired; if the air were admitted to the conditioned space in state X the temperature maintained would be lower than the design value. To avoid this the air must be reheated without change in specific humidity from X to 2.

When reheating is necessary, the refrigeration capacity must be greater than the total heat gain by the amount of heat added during reheating. In this case the temperature of the cooling medium should be as high as is economical in order to get the most refrigerating effect for the expenditure of a given amount of energy. Increasing the temperature of the cooling

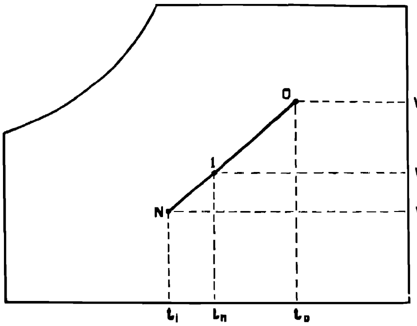

Fig. 3. Mixing of Two Quantities of Air at Different Conditions

medium, however, raises the surface temperature and moves point X to the left, thereby increasing the amount of reheating required, so there is an economical limit to the temperature of the cooling medium used.

c. The **mixing of two quantities of air** in different states may be represented on the psychrometric chart by a straight line joining the points representing the initial states, as shown in **Fig. 3.** Let point O represent the condition of outdoor air. Let point N represent the condition of indoor or recirculated air. The mixture is then represented by state 1 so located that the ratio of the distance from N to 1 to the distance from N to O is equal to the ratio of the supply of outdoor air to the total circulation. Then

$$t_1 - t_N = \frac{M_0}{M}(t_0 - t_N) \qquad (9a)$$

$$W_1 - W_N = \frac{M_0}{M}(W_0 - W_N) \qquad (9b)$$

where M_0 = the weight rate of supply of outside air
M = the weight rate of total circulation (outside plus recirculated air)

d. In **Fig. 4** is shown a summary of what happens to air in **typical summer conditioning.** Air at state 1 is cooled and dehumidified to state 2, where it enters the duct system. In the duct system heat may be transferred to the air, warming it to 3. The air is then introduced to the conditioned space in state 3. Here the air picks up heat and moisture until its state is changed to N. The length of the line from 3 to 4 is proportional to the sensible heat gain of the room (exclusive of the heat gain of outdoor air supplied mechanically and mixed with the return air). The length of the line from 4 to N is proportional to the latent heat gain of the room (exclusive of mechanical outdoor air supply). Air is removed from the room in state N and mixed with outdoor air in state O, thereby increasing its specific humidity and dry-bulb temperature to the conditions represented by state 1. The length of the line from N to 5 is proportional to the sensible heat gain from the supply of outdoor air, and that of the line from 5 to 1 is proportional to the latent heat gain from the supply of outdoor air.

e. Although most of the previous discussion has involved surface cooling, the same principles may be applied when **spray-type** equipment is used. In this case the temperature of the refrigerated spray water would replace the temperature of the cooling surface, t_s, in the analysis of the cooling process.

f. Dehumidification may also be accomplished by using the **absorbing** or **adsorbing properties** of various liquids and solids. Absorbents change physically or chemically during the absorption of water vapor; liquid absorbents include solutions of calcium chloride, lithium chloride, and ethylene glycol, while calcium chloride is also used in the solid form. Adsorbents do not change physically or chemically during the process of dehumidifying air. Solid adsorbents have a porous, sub-microscopic

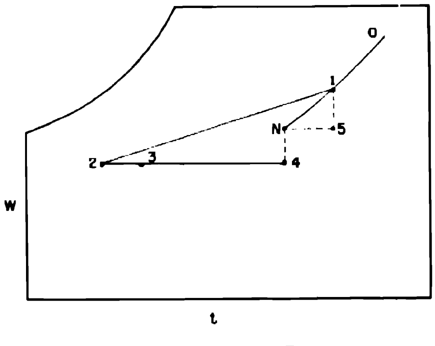

Fig. 4. Summary of Air Passages

structure with many small voids and an extremely large area of adsorbing surface; silica gel, activated alumina, activated charcoal and activated bauxites are examples of solid adsorbents.

Liquid absorbents may be reactivated by heating the solution to drive off water vapor into a stream of activating air. Solid adsorbents are commonly reactivated by contact with a stream of hot (300 F to 400 F) products of combustion or hot air.

In the process of adsorption, the dry bulb temperature of the air increases appreciably and the wet bulb temperature of the air increases slightly as the specific humidity of the air decreases. Typical initial and final states of a mixture of air and water vapor resulting from an adsorption process using silica gel are:

	Initial	*Final*
Dry bulb temperature, F	75	128
Wet bulb temperature, F	70	73.5
Specific humidity, grains per lb of dry air	102.2	38
Enthalpy, Btu per lb of dry air	34.0	36.6

In the above example, the dehumidified, warm air would be cooled before delivery to an air-conditioned enclosure.

Dehumidification by processes of absorption or adsorption is used principally where low humidity must be maintained or where a close control of humidity independent of temperature control is desired.

III. Comfort and Health as Influenced by Air Properties

6. Some of the principal factors that affect human **comfort** insofar as it is influenced by **air environment** are: Dry-bulb temperature, humidity, air motion, air distribution, odors, dust, smoke, bacteria and toxic gases. Other properties of the air that may influence human comfort, but the effects of which are not so clearly established at the present time, are the ozone content and the ionic content. Large changes in air pressure also have effects upon health and comfort that are beyond the scope of this discussion. The simple processes of heating and cooling are used to change the dry-bulb temperature of the air only. Ventilation is the process of changing the air within an enclosure by supplying and distributing fresh air and exhausting used air. Air conditioning, however, is now understood to mean the simultaneous control of at least the first three of the above listed factors.

Human occupancy of an enclosure produces the following **alteration** in the **properties of the air**: (1) oxygen is used in metabolism; (2) carbon dioxide, a waste product of metabolism, is exhaled; (3) the dry-bulb temperature of the air is increased, because the energy liberated in metabolism is partly lost as heat; (4) the humidity of the air is increased by the evaporation of moisture from the body surface and the respiratory tract; (5) odors are given off, and (6) the number of small ions in a unit volume of the air is decreased.

Old and fallacious beliefs were that human discomfort was a consequence of the increase in the **carbon dioxide** content of the air in an enclosure or of the poisonous effect of products of organic decomposition in respired air. The modern theory is that human discomfort results when the temperature-regulating system of the body is unable to compensate comfortably for differences between the rate of energy release in **metabolism** and the rate at which **heat can be dissipated** from the body by evaporation, radiation, convection, and conduction.

7. One arbitrary index of the combined effects of dry bulb temperature, humidity, and air motion upon the feeling of warmth or cold is **effective temperature.** The numerical value assigned to the effective temperature is the dry bulb temperature of saturated still air producing the same feeling of warmth or cold as the air in any given state and motion. Effective temperature, then, is not a temperature at all, but an index that combines the effects of temperature, humidity, and air motion.

The engineering problem involved in providing summer comfort reduces to the maintenance of comfortable and reasonably uniform combinations of dry bulb temperature, relative humidity (or wet bulb temperature), and air motion. At the same time, clean outdoor air must be supplied and vitiated air removed in sufficient quantities to keep by dilution the concentration of odors, smoke, and fumes below objectionable limits. Due to acclimatization or human adaptation to changing air conditions and also to various personal and psychological factors, it is practically impossible to set up invariable standards that will insure comfort for each human occupant in every air-conditioned enclosure. Instead, all that can be done is to prepare standards of recommended practice based upon experience in an attempt to provide comfort for most of the people.

Fig. 1, Chapter 37, is based on A.S.H.V.E. **comfort** data, with lines of constant effective temperature and zones for summer and winter where the majority of people feel comfortable. In the absence of very precise information, it seems desirable to limit the comfort zones to a range of relative humidity from 30 to 70%. This comfort chart applies only to inhabitants of the United States between 20 and 70 years of age living where central heating of the convection type is used during four to eight months of the year. Application of the **summer comfort zone** is further limited to homes, offices, etc., where the exposure of the occupants to the artificial conditions is for periods of 3 hr or more. When the exposure is relatively short, as in a bank or a store, the contrast between indoor and outdoor conditions becomes more evident, and higher indoor temperatures should be carried if the conditioning is to be done for the benefit of the customers. Factors other than the **time of occupancy** which must be used to supplement the information given on the comfort chart are the effects of high or low wall and glass temperatures (affecting radiation to or from the body surface), the state of activity of the occupant (the comfort chart is for people at rest), the amount of clothing worn, the extent of crowding of a room (effect upon radiation from body), and the state of the body at entry to the conditioned enclosure.

8. Table 1 gives suggested indoor conditions that are desirable for certain outdoor dry bulb temperatures; the conditions apply to people engaged in sedentary or light muscular activity where the period of occupancy of the conditioned space is over 40 minutes. This schedule is not absolute and should be used with judgment. For periods of occupancy shorter than 40 minutes, the effective temperatures in the table should be slightly increased. Due to the effects of adaptation, higher effective temperatures may be more comfortable in southern climates.

Table 1. Preferred Schedule of Indoor Air Conditions Corresponding to Outdoor Summer Dry Bulb Temperatures

(Period of occupancy over 40 min; individuals engaged in sedentary or light muscular activity)

Outdoor dry bulb temperature, F	Indoor Air Conditions			
	Effective temperature	Dry bulb, F	Wet bulb, F	Relative humidity, %
100	75	83	66	40
	75	82	67	45
	75	81	68	51
	75	80	70	60
95	74	82	64	36
	74	81	66	44
	74	80	67	51
	74	79	68	57
	74	78	70	68
90	73	81	63	36
	73	80	64	41
	73	79	66	50
	73	78	67	56
84	72	80	61	32
	72	79	63	41
	72	78	64	46
	72	77	66	56
80	71	78	61	36
	71	77	63	45
	71	76	64	52
	71	75	66	61

Constant **air motion** is necessary in an enclosure to maintain uniformity in temperature and humidity. The air distribution may be considered satisfactory if the variation in dry bulb temperature is 3 F or less throughout the frequented portion of a single room. The maximum air velocity in the zone between the floor and the five-foot level that can be tolerated by people at rest is about 50 fpm.

The cooling effect on the clothed human body of air at a given temperature increases with the square root of its velocity; air at 65 F and a velocity of 100 fpm will produce about the same convection heat loss from the clothed human body as air at a temperature of 40 F and a velocity of 15 fpm.

The total **quantity of air** that must be supplied to a conditioned space is fixed by the heat and water vapor loads and the state of the air supplied. Only a part of this air need be introduced from the outside as fresh air, in most cases. The greater the amount of air that can be **recirculated**, the smaller is the load on the refrigerating equipment. In the absence of any other means of removing odors and tobacco smoke, there is a **minimum quantity** of clean outdoor air that must be supplied in a given case to give adequate ventilation. Standards of suggested practice are given in **Table 2.** In no case, however, should the air change be less than that required by local ordinances.

Table 2. Ventilation Standards for Summer Cooling

Application	Cfm per person	
	Preferred	Minimum
Apartment	20	15
Banking space	10	7½
Barber shop	10	7½
Beauty parlor	10	7½
Broker's board room	30	20
Cocktail bar	40	25
Department store	7½	5
Directors' room	40	30
Funeral parlor	10	5
Hospital room	20	15
Hotel room	20	15
Office, general	15	10
Office, private	20	15
Restaurant	15	12
Shop, retail	10	7½
Theater	7½	5
For general application		
Each person, not smoking	7½	5
Each person, smoking	40	25

9. Extended studies have been made and published by the A.S.H.V.E. on the **heat and moisture losses** from the human body at work and at rest under different atmospheric conditions. The human body loses heat by convection to the surrounding air and by radiation to surrounding solid surfaces whenever the temperature of the body is higher than the temperature of the air or of these surfaces. Heat is also dissipated by evaporation of moisture from the body surfaces whenever the temperature of these surfaces exceeds the dew point of the air. The heat lost by radiation and convection increases the sensible cooling load, and the heat lost by evaporation of moisture increases the latent cooling load. The hourly rate of heat loss and the distribution of the heat loss at a dry bulb temperature of the surrounding air of 79 F are given in Fig. 3 of Chapter 37 for several different states of activity.

The total hourly loss of heat from the human body at rest is substantially constant for any effective temperature within the limits of the comfort zones. The distribution of the total loss of heat between sensible heat and latent heat depends chiefly upon the dry bulb temperature of the air and the state of activity. Within the limits of the comfort zones, this distribution is not greatly affected by the humidity of the air, although humidity becomes very important at higher dry bulb temperatures. For a constant state of activity, the fraction of the total loss that is in the form of sensible heat decreases as the dry-bulb temperature of the air increases. The effect of dry bulb temperature upon heat and moisture losses from the human body is shown in **Table 3.**

10. a. The **quality** of air, from the standpoint of ventilation, is affected by the amount and nature of the **dust** carried. Dust is sometimes classified as solid particles in the air between 1 and 150 microns in diameter (one micron is one-thousandth of a millimeter). Classified in this manner, dusts settle in accordance with the laws of gravity. Some industrial operations produce dusts that, due either to their abrasive or toxic actions, are injurious to health when inhaled.

The **dust content** of the air may be measured by impingement, scrubbing, electrostatic precipitation, or settling. In the Owens dust counter, a measured volume of air is drawn into the apparatus by a piston, moistened and caused to impinge upon a

Table 3. Heat Losses from Human Body

A. As affected by state of activity; indoor dry bulb temperature of 80 F (ACRMA Application Engineering Standards, 1947)

Activity	Total loss, Btu per hr	Sensible heat loss, Btu per hr	Equivalent latent heat loss, Btu per hr	Moisture evaporated, grains per hr
Seated at rest	330	180	150	1,000
Seated, very light work	400	195	205	1,370
Moderately active, standing, light work	450	200	250	1,670
Light factory work	750	220	530	3,560
Moderately heavy factory work	1,000	300	700	4,660
Heavy work	1,450	465	985	6,560

B. As affected by room dry bulb temperature

Indoor dry-bulb temp F	Per person seated at very light work		Per person at light factory work	
	Sensible, Btu per hr	Equivalent latent, Btu per hr	Sensible, Btu per hr	Equivalent latent, Btu per hr
80	195	205	220	530
75	240	160	295	455
70	275	125	365	385

Table 4. Physiological Effects of Gases and Vapors

(Concentration in parts per 10,000 parts of air by volume)

Substance	Rapidly fatal	Maximum allowable for prolonged exposure
Ammonia	50	1
Aniline	—	0.05
Arsine	2.5	—
Benzene	200	1
Carbon dioxide	1000	50
Carbon disulfide	20	0.2
Carbon monoxide	40	1
Chlorine	10	0.01
Gasoline	240	5
Hydrochloric acid	10	0.1
Hydrocyanic acid	30	0.2
Hydrofluoric acid	2	0.03
Hydrogen sulfide	6	0.2
Lead	—	5
Nitrobenzene	—	0.01
Phosgene	0.5	0.01
Phosphine	10	—
Sulfur dioxide	4	0.1
Toluene	200	2
Turpentine	—	1
Xylene	200	2

glass disc. The dust particles collected on the disc are counted under a microscope. One arbitrary rule concerning the quality of the air is that the number of dust particles present in a well-ventilated room should not exceed 25,000 per cu ft.

b. Fumes are usually classified as particles between 0.2 and 1 micron in diameter and may result from distillation or chemical reaction. The physiological effects of certain gases and vapors appear in **Table 4.**

c. Smoke may be classified as particles less than 0.3 micron in diameter, usually resulting from incomplete combustion. Smokes do not settle from the air and may be so fine as to penetrate, in part, most filters. To ventilate a room occupied by tobacco smokers may require a supply of as much as 20 cfm of air per smoker more than is supplied when the occupants are not smoking.

d. The usual procedure in determining the **bacteria** content of the air is to expose culture plates to the air for 2 min, then incubate at 98 F for from 24 to 48 hr and count the colonies of bacteria which have developed under a microscope.

11. Distinct from the effect of the properties of air upon human comfort are the environmental influences upon **health.** Such influences are difficult to measure due to inability to establish a concise and accurate definition of health or disease. The general relationship between extreme conditions of weather and human health is well known. High environmental temperatures and humidities produce a rise in body temperature owing to a decrease in the rate of heat loss by radiation, convection, and evaporation. When the temperature of the environment attains that of the body, the loss of heat by radiation and convection stops.

However, the effectiveness of air conditioning in the treatment of individuals suffering from symptoms of **hay fever** and **pollen asthma** has been demonstrated. Also of interest but of no statistical value, because of the limited number of subjects, are the experiments conducted by Kerr and Lagen, which seem to indicate that the **common cold** cannot be transmitted from one person to another in a properly air conditioned room.

A high relative humidity (about 65%) has been found to be beneficial to the health and growth of premature infants up to the time that the infants reach a weight of about 5 lb.

Airborne contagion may be reduced by dust elimination and droplet-nuclei disinfection. Solutions of triethylene glycol and propylene glycol are germicidal for most airborne bacteria, particularly in dust-free air with relative humidity between 40 and 60%. Additional data, however, are needed for conclusive evidence of the effects of glycol solutions and ultraviolet radiation.

Part II. Air Purification*

Purification of air is one of the major functions of air conditioning. Just as air temperature may be controlled through the use of heating and cooling coils in the system, so the content of air as regards dust, fumes, pollens, odors and other contaminants may be controlled through the use of the various air cleaning devices which are available. These devices are installed in the duct system, preferably ahead of the coils or at the intakes of unit air conditioners. Sufficient area is provided to assure the air velocity at which the particular device operates with the greatest efficiency. Provision is made for the removal of the filtering elements for cleaning or reactivation. In many installations, permanent gages are installed which indicate the resistance through the cleaning device. When the resistance reaches a prescribed point, the elements are taken out and cleaned or thrown away according to the type of equipment used.

In practically all air conditioning systems, a certain quantity of air is discharged from the system and replaced with **fresh air** or new air. The balance of the air is recirculated through the conditioner, is recooled, reheated, humidified or dehumidified according to what function the conditioner may be performing at the time.

In some systems where the contamination of air is very slight, the recirculated air is not cleaned but is mixed with the fresh air which is subjected to a cleaning treatment before it enters the system. In other cases, where contamination is heavy, both fresh and recirculated air are treated. In certain systems, even the air wasted must be cleaned before it may be discharged to the atmosphere because it must always be remembered that what may be "waste air" to one system may become the "fresh air," in part or in whole, of a system installed next door.

Fresh air, so-called, is air taken into the system from out-of-doors to make up for the air discharged from the system. Its degree of freshness may vary widely. It may contain very little contamination or it may well be less "fresh" than the air in the conditioned area and so may require a great deal of cleaning or purification. Air recirculated in the conditioned space may pick up anything from a little fine dust to heavy, airborne waste from machine processes, smoke, fumes, organic vapors and odors. The amount of purification required in any given system must, therefore, be based on the job the system has to do, the conditions under which it must operate and the content of the air available for makeup.

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* This portion of Chapter 35 was prepared by C. N. Deverall.

The selection of **equipment** for the system becomes the problem of the designing engineer, who must evaluate all of the points mentioned and decide what type of equipment or what combination of types of equipment will best suit his needs. An expensive system which might be perfectly justified in one case might be entirely out of line in another, from the standpoint of first cost, operation and maintenance.

Equipment Types

There are a number of methods available for the cleaning and purification of air. All of them have certain merits; most of them have certain deficiencies. Some of the methods will remove heavy particles but will pass the fine particles. Some will efficiently collect the fine particles but would soon become clogged if subjected to the heavy particles. Some will remove fumes, organic vapors or odors but would not be suitable for the removal of solids. Very frequently a combination of two or more of the above-mentioned types will be the answer.

The various types of equipment available will fall into the following general classifications:

Centrifugal collectors
Air washers
Dry-type filters
Viscous filters for manual operation
Viscous filters for automatic operation
Electrostatic precipitators
Adsorption equipment

The **centrifugal collector** will be effective in the removal of the heavy, airborne contaminants found in industrial plants, such as waste from woodworking machines, exhaust from standing, grinding and buffing operations, fly ash, etc., all of which have particles of considerable magnitude and specific weight. These heavy particles are "whirled out" of the air stream as it flows through a centrifugal collector operating on the centrifuge principle. After the air has passed through this device, it is either wasted to the atmosphere or taken through a secondary process composed of equipment of a different type which will remove the finer particles, rendering the air fit for reuse.

Air washers fall into two general types. One is the **spray washer** composed of a watertight casing, banks of spray nozzles which break the water up into very fine droplets and eliminators for the arresting of entrained moisture which would otherwise go through into the air stream. The other type is the **surface or scrubber type** which presents a very considerable surface of scrubber plates to the air stream, which surface is sprayed or washed down constantly with spray nozzles from above. Impurities are removed in the spray unit through the action of the falling droplets from the sprays which contact the contaminants and carry them down into the pan. In the scrubber unit, the air passing through the unit is impinged upon the wetted surface, the contaminants captured and carried down through the washing action of the sprays above.

The air-cleaning function in air washers is usually combined with the treatment of air to control its temperature and moisture content. As a matter of fact, the cleaning action is rather secondary to these other two important functions. The control of the spray temperature permits delivery of air at a given dew point which, in turn, controls the moisture content of the air in the system. It does, however, do a pretty good job of cleaning where certain types of impurities are present. In the case of water-soluble, organic vapors a pretty fair job of cleaning can be done.

The scrubber principle is used in certain types of waste-air systems, such as paint-spray booths, etc., where the air is drawn through vertical eliminators which are washed down with scrubber nozzles. In many localities, paint-spray waste would not be permitted to be discharged to the atmosphere without first going through some kind of a cleaning system.

Dry-type filters, so-called, are either of the throwaway type or the semi-permanent type. The throwaway type filter, as its name indicates, is made up of a number of elements which, after they have picked up their quota of airborne impurities, are discarded and replaced with new ones. They are usually made up of cardboard frames filled with filter media of paper, gauze, metallic or glass wool, animal hair, etc. They are necessarily made of such materials that their cost will be low. The semi-permanent types are made up in the same general form but are provided with permanent metal frames and supports for the filter media. When they have performed their function, they are withdrawn from the frame; the filter media is removed, thrown away and replaced by the user. With this type of filter, the permanent sections will be built of durable materials but the filter media must, of course, be of low cost.

Viscous-type filters of the manually-operated type are usually made up in much the same form as the dry-type filter as regards to size, etc. In this type of filter, however, the frame and filter media are of a permanent type and are considerably more expensive. The filter media is usually made of durable wools or fibres, screen wire, metal punchings, etc., contained in a metal frame and treated with a viscous oil which captures and holds the particles picked up through impingement in their passage through the filter. When the filter becomes loaded to the point where further resistance would unduly retard air flow, it is taken out of the bank and cleaned in an oil bath, allowed to drain and replaced in service.

Both the dry-type and manually-operated viscous filters are usually mounted in permanent slides placed in the intake air ducts. Access must be provided for the removal of the filter elements either by pulling them through slides in the wall of the ducts or by entering the duct through an access door and removing them from the face of the frame.

Automatic viscous filters are usually made up of frames somewhat similar to those described for manually-operated viscous filters or of a series of perforated metal plates through which the air must pass. The frames or plates are mounted on an endless chain and are driven in vertical motion through the air stream, then through a cleaning bath or trough in the bottom of the unit. The plates or frames pass through this cleaning bath, are cleansed of their impurities, recoated with the adhesive and drained before they reenter the air stream. This type of filter is very efficient, is very widely used and is, of course, correspondingly expensive from a first cost standpoint. Where a large amount of air is handled the year around, the comparatively low service cost of this type of filter should be considered.

The **electrostatic precipitator** operates on the principle that airborne particles subjected to the influence of a high-tension electric field will become charged or ionized and will be precipitated when they contact an electrode of opposite polarity. The units are made up of a series of plates, coated with a viscous material which captures and holds the precipitated particles.

In one type of unit which is manually operated a periodic shutdown is necessary for the cleaning of the device. The plates are washed down with water sprays and then recoated with the viscous oil. In another type plates are mounted on an endless chain very similar to the arrangement described for the automatic viscous filter, and the plates are carried successively through the air stream, then through a cleaning bath in which the dust particles are removed and the surface recoated for another trip through the air stream.

The electrostatic precipitator has found wide use in the removal of heavier, airborne particles such as fly ash, etc. It is frequently used in conjunction with dry or viscous filters as the first stage, taking out the heavier particles, after which the air passes through the dry or viscous filters for final cleaning.

Adsorption equipment, as the name implies, is used for the adsorption of odors, organic gases and vapors. The air is usually taken through a first stage filter before passing on to this device. There are a number of adsorbents, of which activated charcoal or activated carbon is a good example. This material has a very high adsorption value, as it will pick up 40% to 50% of its own weight in many of the gases ordinarily found in the air we breathe. By properly proportioning the adsorptive area, the system may be made to operate for long periods of time without servicing. When they become saturated the filter beds are reactivated by the application of heat at about 1,000 F.

Activated carbon is placed in perforated containers or canisters, so arranged that they may be placed in the air stream and easily removed for reactivation. The time cycle of reactivation will depend upon the retentivity of carbon for the particular gases or vapors picked up in this operation. This type of equipment has been found particularly desirable in the removal of odors in rooms used for public assembly, such as restaurants, theatres, meeting halls, etc. Much commendable work has been done in the design of the equipment for ready servicing. Application Data Section 42, published in the September, 1948, issue of *Refrigerating Engineering*, covers this subject in detail.

Design Data

No attempt is made here to present design data. The manufacturers of the various types of equipment have done a great deal of research, each on his own particular equipment. These manufacturers will be glad to supply very complete design data upon request and, as improvements are constantly being made in the equipment, it is well to keep in touch with these manufacturers so that the very latest information may be available.

A certain amount of similarity exists in the respective types and considerable physical standardization has been done in the way of element sizes and general installation arrangements. Filters 16 in. × 20 in., 20 in. × 20 in., 16 in. × 25 in. face area will be found in many of the manufacturers' lines and, indeed, a great many of the filters are interchangeable with other makes of filters for a given frame arrangement. The thickness of these filter elements varies from 1 in. to 4 in. or 5 in. Two 1-in. elements are frequently used, the intake element only being discarded at each cleaning. The second element, being fairly clean, is brought forward to take the place of the discarded section and a new one inserted back of it.

Air Velocity

Air velocities through air cleaning equipment will vary to a considerable degree. In centrifugal equipment, velocities will range from 50 to 2,000 fpm, according to the specific weight and nature of the particles handled. Air washer velocities will average around 500 fpm. Velocities through filters of the dry type seem to average around 300 fpm, while in the viscous filters they will vary from 300 to 500 fpm. Air resistance data are given in terms of clean filters and will vary in the different makes from .06 to .2 in. H_2O, the higher resistance as a rule existing in the filters in which higher air velocities are used.

The manufacturer should be consulted as to the arrangement of the equipment and should be fully acquainted with the specified design data as to allowable air resistance, cfm, etc. Local safety codes and underwriters' requirements should be carefully studied before the choice of any filter is made. Good practice would indicate filter media of a fire-resisting type and it will be found that many codes specifically require this feature.

In general, the choice of the equipment either in individual types or in combination should be made upon the basis of the work to be done, the efficiency desired and the conditions under which the equipment must operate.

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36. FANS AND DUCTS

I. Fans

FANS are used in air conditioning and ventilation applications to produce air flow at free delivery or through duct systems. The type and style of fan to be used depends upon the noise limitations, volume of air required, and the static pressure of the given application.

1. The usual **classifications** are the propeller type and centrifugal type. The **propeller fan** is used where air movement against small static pressures is required, and usually affords quieter operation at low pressures than the **centrifugal fan.** This is usually in cases where little or no ductwork is used. The centrifugal fan will operate efficiently against the higher static pressures encountered in duct systems, and with less noise than the propeller fan. Central plant air conditioning systems and ventilating systems using ducts for air distribution generally use centrifugal fans ("blowers") because the static pressures encountered are usually too high to be handled efficiently by a propeller fan.

In the propeller fan, air is discharged in a direction parallel with the shaft. In the case of the centrifugal fan, the blades may be straight, as in a paddle wheel, or curved forward or backward in the direction of air flow. Air is discharged from centrifugal fans in the plane of rotation. Many designs of both types are on the market.

Centrifugal fans with **forward curved blades** differ in performance from those with **backward curved blades.** To yield a specified air output the former have to be run at slower speed, or they may be smaller in diameter. If similar fans are run at the same speed, the output of the one with forward curved blades will be considerably greater than that of the fan with backward curved blades.

There is also a difference in typical characteristic performance as is shown in **Figs. 1, 2 and 3.**

Sometimes it is desirable to adopt suitable **fan speeds** in relation to output, which may work out to better advantage with one style wheel than with the other.

A third style of centrifugal fan utilizes wide **radial blades.** Such fans are often used in very large sizes.

Total pressure of a fan is the difference between the total pressure head (sum of static and velocity pressure heads) of the air leaving, and the total pressure head of air entering the fan. Total pressure is thus a measure of the useful work done on the air.

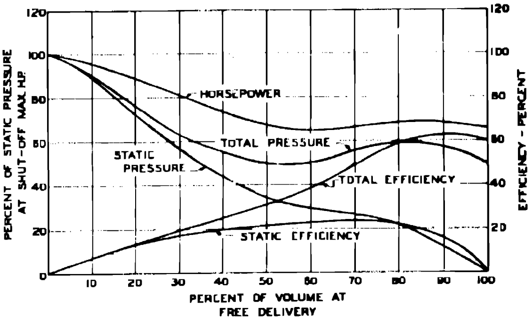

Fig. 1. Typical Characteristics of Propeller Fans

Static pressure is the pressure of air at rest, or it is the pressure exerted equally in all directions independently of any pressure exerted by the air because of its motion.

Static pressure of a fan may be defined arbitrarily as the total pressure of the fan minus the velocity pressure corresponding to the air velocity at the fan outlet. This is most commonly accepted as a definition of static pressure at the present time, but, though useful, it is not a true definition. Since it is an arbitrary statement, it may be subject to modification, and in working with fan data care should be exercised to

ascertain the exact definition used in connection with such data.

Air horsepower is a measure of the work required to move the actual volume of air against a pressure equal to the total fan pressure, thus:

$$\text{Air horsepower} = \frac{\text{cfm} \times (\text{total pressure in in. } H_2O)}{6{,}356}$$

Standard air is air weighing 0.075 lb per cu ft, which corresponds to air having a barometric pressure of 29.921 in. Hg, a dry-bulb temperature of 68F and 10% relative humidity. This definition is universally accepted and gives a standard basis for all air flow calculations. Laws which will be given later may be used for corrections when air deviates from standard conditions.

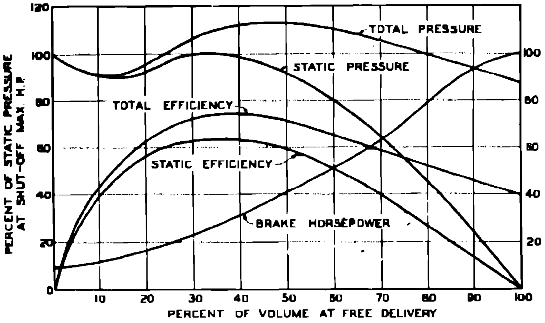

Fig. 2 (Left). Characteristic Curves for Forward Curved Centrifugal Fans

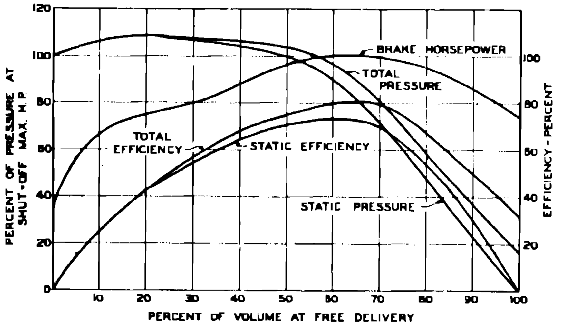

Fig. 3 (Right). Characteristic Curves for Backward Curved Centrifugal Fans

Total efficiency, also called mechanical efficiency, is the ratio of the power required to move the actual volume of air against the total pressure of the fan to the shaft power input (brake horsepower):

$$\text{Total efficiency} = \frac{\text{cfm} \times (\text{total pressure in in. } H_2O)}{6{,}356 \times (\text{hp input to shaft})}$$

Static efficiency is the ratio of the power required to move the actual volume of air against the static pressure of the fan to the shaft input:

$$\text{Static efficiency} = \frac{\text{cfm} \times (\text{static pressure in in. } H_2O)}{6\ 356 \times (\text{hp input to shaft})}$$

Brake horsepower is the input to the fan shaft, but does not always include bearing or belt losses. This point must be watched carefully when selecting the driving motor.

2. Fan characteristic curves, representative of propeller fans in general, are shown in Fig. 1. The curves show that the capacity of the fan drops quickly with an increase of static pressure, and that the horsepower increases because of the reduced efficiency with increased pressure. Hence, propeller fans are limited, for the most part, to the movement of relatively large volumes of air at comparatively low static pressures.

Characteristic curves for a **forward curved** centrifugal fan are shown in Fig. 2. This type of fan is widely used, has a large capacity for the space occupied, and is quiet in operation. The point of maximum efficiency is very close to the point of maximum static pressure. Forward curved fans should be operated close to this point. At lower pressures operation may be accompanied with objectionable noise. Beyond the point of maximum efficiency (to the left of the peak of the curve of static efficiency), forward curved fans may be unstable in operation and subject to wide variations in air flow with comparatively small changes in static pressure. This may be accompanied by excessive noise caused by fluttering, or sudden changes in air flow. The power curve rises continuously from a minimum at shut-off to a maximum at free delivery, showing that from the standpoint of the fan alone the driving motor might be subject to overloading or underloading in case it were subject to fluctuating static pressure.

Fig. 3 shows characteristic curves for a **backward curved** centrifugal fan. Its usual high-speed operation often makes desirable direct-connected motor drive, even with large size fans, and its non-overloading power curve has advantages. The steep pressure curves to the right of the point of maximum efficiency give a relatively flat regulation curve, so that considerable changes in static pressure do not greatly affect the air flow. The maximum power requirement generally occurs at the point of maximum efficiency. If a motor is selected for this point, the possibility of its being overloaded is eliminated, even though the static pressure of the system is considerably smaller than expected.

There are other variations in centrifugal fans between the full forward and full backward curve types, but they all have characteristics varying between those of the above mentioned fans.

3. The actual operation of a fan in a given system depends upon the fan characteristics and the **system characteristics. Fig. 4** shows how a given fan will operate with a typical duct system. The fan will always operate at the point of intersection of the system characteristic with the fan characteristic. When a different operating condition is required, it is necessary to change either the fan speed or the system characteristic. Referring to this figure, suppose a fan is operating at point C, the intersection of system II characteristic with the 265 rpm fan characteristic, where it is delivering 21,400 cfm. If it is desired to reduce the air flow to 19,000 cfm it can be accomplished by changing the fan speed to 235 rpm, in which case the point of operation will be point A. Another method is to throttle the duct system so that the point of operation is D. When the desired result is produced by changing the fan speed, the operating point moves along the system characteristic and remains at the same high efficiency point on the fan characteristic. When the change is made by throttling, the operating point moves along the fan characteristic curve toward the unstable region, with increased static pressure and some decrease in required motor horsepower. However, this decrease in motor horsepower is less than if the change is effected by reducing the fan speed, because the efficiency is decreased.

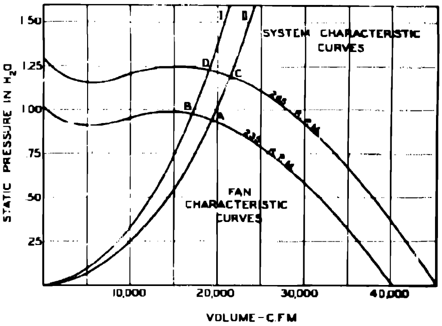

Fig. 4. System Characteristic Curves and Fan Characteristic Curves for a Forward Curved Centrifugal Fan

Changes in Operating Characteristics

4. The following principles may be used to determine the operating characteristics of fans, either centrifugal or propeller, when the conditions of operation change. These laws, although approximate, are sufficiently accurate for most practical purposes.

- a. With a given fan in a given duct system and constant air density:
  - 1. The capacity varies directly as the fan speed.
  - 2. The pressure (either static or total) varies directly as the square of the fan speed.
  - 3. The fan horsepower varies directly as the cube of the fan speed.
- b. With a constant weight of air circulated by the fan:
  - 1. The speed, capacity, and pressure vary inversely as the density.
  - 2. The horsepower varies inversely as the square of the density.
- c. To maintain a constant pressure at the fan, the speed, capacity, and horsepower must vary inversely as the square root of the density.
- d. With speed constant and capacity constant, the pressure and horsepower vary directly as the density of the air.

5. The first of the above rules is useful in determining the effect of a change of speed on the operating characteristics of a fan. The others are useful in selecting fans and determining their performance when the air density differs appreciably from standard because of change of temperature or elevation above sea level. The following **examples** are illustrative of their use.

I. A certain fan is operating in a given system under the following conditions, with 68F air.

Capacity = 19,000 cfm
Static pressure = 0.93 in. H_2O
Speed = 235 rpm
Bhp = 4.9

What will be the conditions of operation if the speed of the fan is increased to 265 rpm without making any changes in the distribution system? From rule a:

$$\text{Capacity} = 19{,}000 \times \left(\frac{265}{235}\right) = 21{,}400 \text{ cfm}$$

$$\text{Static pressure} = 0.93 \times \left(\frac{265}{235}\right)^2 = 1.18 \text{ in. } H_2O$$

$$\text{Horsepower} = 4.9 \times \left(\frac{265}{235}\right)^3 = 7.0$$

These laws may be applied to the calculation of new fan characteristic curves for different speeds, if the characteristic curve for one speed is known. In the case of Fig. 4, suppose that it is required to determine the location of the static pressure curve for 265 rpm, when the 235 rpm curve is known. The expressions for finding the point D corresponding to the point B are as follows:

$$\text{Static pressure} = 0.96 \times \left(\frac{265}{235}\right)^2 = 1.22$$

$$\text{Cfm} = 16{,}800 \times \left(\frac{265}{235}\right) = 19.000$$

Therefore D is definitely located by the above values of static pressure and cfm. In like manner, other points may be located and the complete curve plotted.

II. A certain fan application has the following requirements:

Required cfm = 10,000 at 68F
Static pressure = 0.59 in. H_2O
Mean barometric pressure = 25.4 in. Hg.

The problem is to select the fan and determine its speed and horsepower:

- a. The air density at 25.4 in. Hg

$$= 0.075 \times \left(\frac{25.4}{29.92}\right) = 0.06357$$

- b. Use rule b to work backward to determine the cfm and static pressure, with standard air, to be used in selecting the fan from standard rating tables.
  - 1. The volume of standard air equivalent to the same mass of required air is

$$10{,}000 \times \frac{0.06357}{0.075} = 8{,}500 \text{ cfm}$$

  - 2. The static pressure varies inversely as the density.

$$0.59 \times \frac{0.06357}{0.075} = 0.50$$

- c. The following data are obtained from a standard rating table:
  Capacity = 8,500 cfm
  Static pressure = 0.50 in. H_2O
  Speed = 207 rpm
  Brake horsepower = 1.07 hp
- d. Use rule b to check the fan performance with 68F air at a barometric pressure of 25.4 in. Hg.

$$\text{Speed} = 207 \times \frac{0.075}{0.06357} = 244 \text{ rpm}$$

$$\text{Capacity} = 8{,}500 \times \frac{0.075}{0.06357} = 10{,}000 \text{ cfm}$$

$$\text{Static pressure} = 0.50 \times \frac{0.075}{0.06357} = 0.59 \text{ in. } H_2O$$

$$\text{Bhp} = 1.07 \times \left(\frac{0.075}{0.06357}\right)^2 = 1.48 \text{ hp}$$

III. Take the case of a certain fan delivering 3,000 cfm at 230 rpm, with 4.6 hp, at an air temperature of 68F (density=0.075) and a static pressure of 1.00 in. H_2O. Determine the speed, capacity, and horsepower requirements if the pressure is maintained constant and the air temperature rises to 220 F (density 0.0584). From rule c:

$$\text{Speed} = 230\sqrt{\frac{.075}{.0584}} = 261 \text{ rpm}$$

$$\text{Capacity} = 3{,}000\sqrt{\frac{.075}{.0584}} = 3400 \text{ cfm}$$

$$\text{Bhp} = 4.6\sqrt{\frac{.075}{.0584}} = 5.21 \text{ hp}$$

IV. An air conditioning system located in a city where the mean barometric pressure is 25.4 in. Hg (air density at 68F =0.06357 lb per cu ft) requires that 10,000 cfm of 68F air be circulated. The static pressure under actual operating conditions is calculated as 0.59 in. H_2O. It is required to select a fan, determine its speed and horsepower.

a. Use rule d to work backward to determine the cfm and static pressure, with standard air, to be used in selecting the fan from standard rating tables.

The capacity will remain constant at 10,000 cfm.

The static pressure will vary directly as the density.

$$\text{Static pressure with standard air} = \frac{0.075}{0.06357} \times 0.59 = 0.695$$

b. The following data are obtained from a standard rating table.

Capacity = 10,000 cfm

Static pressure = 0.695 in. H_2O

Speed = 244 rpm

Brake horsepower = 1.74 hp

c. This fan will have the following characteristics, with 68 F air at 25.4 in. barometric pressure (by rule d):

Speed = 244 rpm

Capacity = 10,000 cfm

$$\text{Static pressure} = .695\left(\frac{0.06357}{0.075}\right) = 0.59 \text{ in. } H_2O$$

$$\text{Bhp} = 1.74\left(\frac{0.06357}{0.075}\right) = 1.48 \text{ hp}$$

Fan Selection

6. In order to select the proper fan for a given installation it is necessary to know the following:

a. Volume of air required per minute at a definite density
b. Total resistance pressure of the system when the fan is delivering air of this same density
c. Noise limitations
d. Space available
e. Type of load, variable or otherwise
f. Type of motive power available

Fan manufacturers simplify fan selection by supplying tables or curves of fan characteristics, which usually give the following information for each size of fan:

a. Volume of air handled
b. Outlet velocity
c. Fan speed in rpm
d. Tip or peripheral speed of fan rotor
e. Brake horsepower
f. Static pressure

This information is given for air under standard conditions, i.e., 68 F, 10% relative humidity (density=0.075 lb per cu ft).

Fans should be operated as closely as possible to the point of maximum efficiency. This also usually corresponds to the point of quietest operation.

Table 1 gives representative values of outlet velocities and tip speeds for forward and backward curved centrifugal fans. Tables of this sort are an aid in selecting fans which will operate near their points of maximum efficiency and with a reasonable degree of quietness. Although the numbers

Table 1. Air Velocities and Tip Speeds Used with Multiblade Fans

Press (static) in. H_2O	Forward curved fans		Backward curved fans	
	Outlet vel, fpm	Tip speed, fpm	Outlet vel, fpm	Tip speed, fpm
0.250	1,000-1,100	1,500-1,700	1,000-1,100	3,300-3,700
0.375	1,000-1,100	1,700-1,900	1,000-1,100	3,700-4,100
0.500	1,000-1,200	1,900-2,100	1,100-1,300	4,100-4,500
0.625	1,100-1,300	2,100-2,400	1,100-1,300	4,500-4,700
0.750	1,200-1,400	2,400-2,700	1,100-1,300	4,700-4,900
0.875	1,300-1,600	2,700-2,900	1,100-1,200	4,700-4,900
1.00	1,400-1,800	2,900-3,100	1,000-1,100	4,900-5,100
1.25	1,700-1,900	3,100-3,400	1,000-1,100	5,000-5,200
1.50	1,800-2,100	3,400-3,800	1,000-1,100	5,200-5,600
1.75	1,900-2,300	3,800-4,200	900-1,000	5,600-6,000
2.00	2,200-2,600	4,200-4,500		
2.25	2,400-2,800	4,500-4,700		
2.50	2,600-3,000	4,700-4,900		

here are representative of the general run of fans of these types, values recommended by the manufacturer of the particular fan under consideration should be used.

The following precautions should be observed when applying fans to air conditioning systems. The fan should be selected to deliver the required cfm at a slightly greater static pressure than the calculated value of the system, in order to allow for small errors in calculation and to assure the delivery of the required amount of air. Also, the motor should have a nameplate rating somewhat in excess of the brake horsepower as obtained from the fan catalog data, for the following reasons:

a. To provide for losses in the bearings of the fan, and belt drives, when used.

b. To provide a margin for increasing the fan speed in case it is found, after installation, that the fan will not deliver the required output.

c. To provide a margin of reserve in the case of a forward curved fan to prevent overloading the motor in case anything happens to decrease the resistance pressure of the system, or in case the calculations were too high.

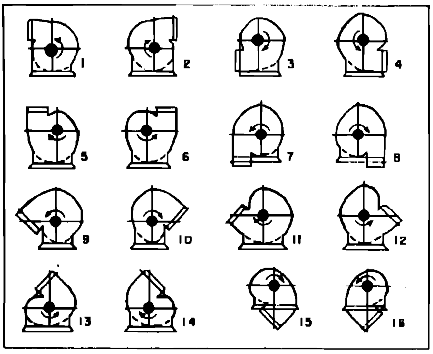

Fig. 5. Standard Designation of Fans

1. Counter-Clockwise Top Horizontal. 2. Clockwise Top Horizontal. 3. Clockwise Bottom Horizontal. 4. Counter-Clockwise Bottom Horizontal.

5. Clockwise Up Blast. 6. Counter-Clockwise Up Blast. 7. Counter-Clockwise Down Blast. 8. Clockwise Down Blast.

9. Counter-Clockwise Top Angular Down. 10. Clockwise Top Angular Down. 11. Clockwise Bottom Angular Up. 12. Counter-Clockwise Bottom Angular Up.

13. Counter-Clockwise Top Angular Up. 14. Clockwise Top Angular Up. 15. Clockwise Bottom Angular Down. 16. Counter-Clockwise Bottom Angular Down.

The mechanical arrangement of a fan is important, and is influenced by the space available and the arrangement of the system. The National Association of Fan Manufacturers[1] has adopted a method of designating fans as follows (Fig. 5): Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is counter-clockwise, the designation is **counter-clockwise.** (The driving side of a single inlet fan is considered to be the side opposite the inlet, regardless of the actual location of the drive.) This method of designation applies to all centrifugal fans, single or double width, and single or double inlet.

The discharge of a fan will be determined by the direction of the line of air

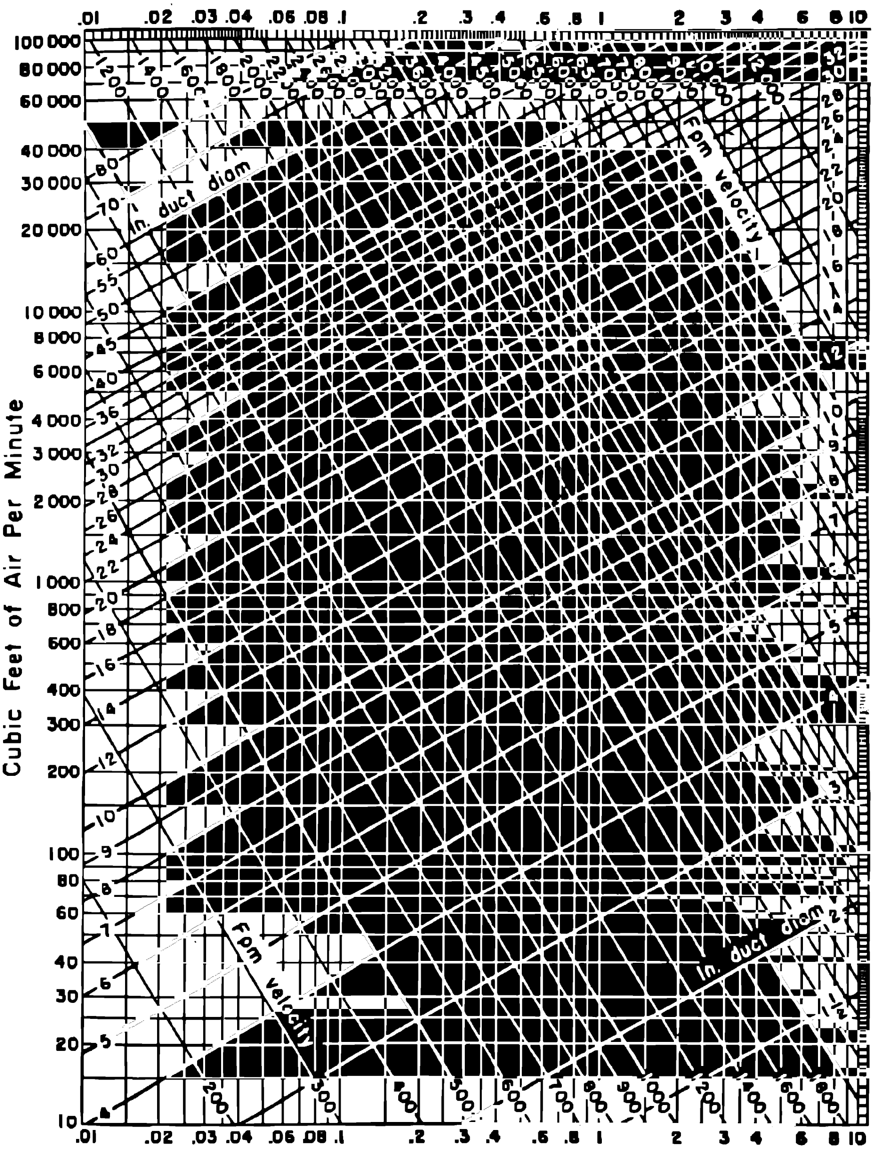

Fig. 6. Friction Losses in Round Ductwork

discharge and its relation to the fan shaft, as:

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of discharge is vertically upward.

Down blast: If the line of discharge is vertically downward.

In addition to the designations as to discharge, fans are supplied with different arrangements, and single or double width, single or doublet inlet. A single inlet fan takes in air from only one side; a double inlet fan from both sides. Double inlet fans are usually approximately twice as wide as single inlet fans.

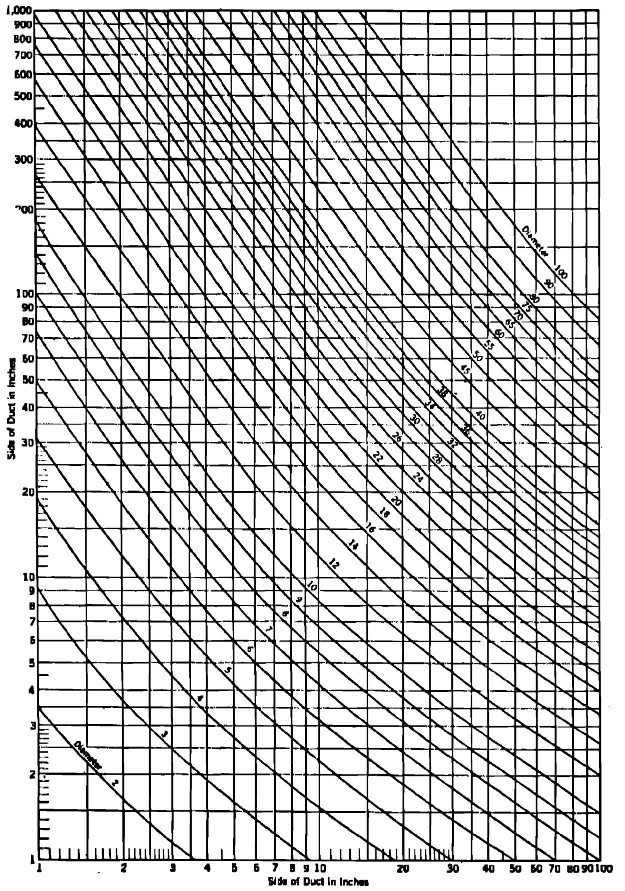

Fig. 7. Equivalent Diameters of Rectangular Ducts

II. Design of Duct Systems

This section discusses the design of a duct system from the standpoint of determining the correct sizes of ducts to carry the right amount of air to each supply opening or from each return opening, and of determining the pressure drop in the duct system. Hence, a considerable portion of this section is devoted to the pressure drops in various parts of a duct system. A knowledge of what causes pressure loss is of value not only in determining the total pressure drop of a given system, but also in avoiding arrangements which cause excessive pressure loss, and in balancing pressure losses in the various branches to proportion air flow properly.

7. The **pressure drop** in a duct system is made up of two components:

a. Frictional resistance between the air and the duct walls.

b. Dynamic losses occurring at pipe entrances, elbows, takeoffs, grilles and other points where the velocity or direction of flow changes.

Friction losses may be expressed by the usual hydraulic equation for fluid flow. For practical engineering purposes this has been converted to the form of **Fig. 6.** This chart gives friction loss in inches of water per 100 ft of straight duct having a circular cross section vs. cfm air velocity, and duct diameter. It is based on standard air.

As an example, it is desired to convey 2,000 cfm of standard air through a duct with a resistance not to exceed 0.06 in. per 100 ft. From the chart it is found that a 20 in. round duct at 900 fpm may be used.

Fig. 7 gives the sizes of rectangular and round ducts offering equal frictional resistance when the same amount of air flows through each. As an example, a 22 in. round duct has essentially the same frictional resistance as a 20×20 in. square duct of a 10×45 in. rectangular duct.

It should be noted that for the same quantity of air the velocity in a rectangular duct is somewhat lower than that in an equivalent round duct.

The pressure loss through a given rectangular elbow varies according to its shape.

8. **Dynamic losses** are usually expressed as a certain number of velocity heads. Velocity head is given by Equation (1)

$$H_v = \frac{v^2}{2g} \qquad (1$$

where

H_v = velocity head, ft of air
v = velocity, fps
g = 32.2 (acceleration of gravity)

Pressures in air conditioning systems are commonly measured in in. H_2O, and velocities in fpm. Equation (1) thus becomes:

$$h_v = \frac{(V/60)^2}{2g}\,\frac{12w}{62.4}$$

$$_v = \left(\frac{V}{1097}\right)^2 w \qquad (2)$$

where

h_v = velocity head, in. H_2O
V = velocity, fpm
w = density of air, lb per cu ft

For standard air this reduces to

$$h_v = \left(\frac{V}{4005}\right)^2 \qquad (3)$$

9. Pressure losses in **90 deg elbows** may be determined from **Figs. 8** and **11.** Figs. 8, 9, and **10** give the per cent velocity head lost vs. the ratio of the radius of curvature of the center line of the elbow to the depth of the duct. The solid curves are for elbows followed by a section of straight pipe several diameters in length. The dotted curves are for elbows discharging directly.

Pressure losses **in round elbows** are given in Fig. 8. The solid curve is taken directly from Bussey[2], while the dotted curve is calculated from the other by means of the data for pipe regain given by Wirt[3].

Pressure losses **in square elbows** are given in Figs. 9 and 10. These are taken from Wirt[3], who has given most satisfactory compilation of data on losses in rectangular elbows.

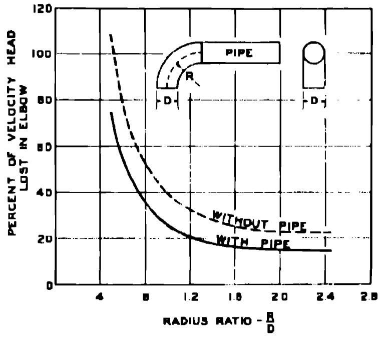

Fig. 8. Resistance of Round Elbows

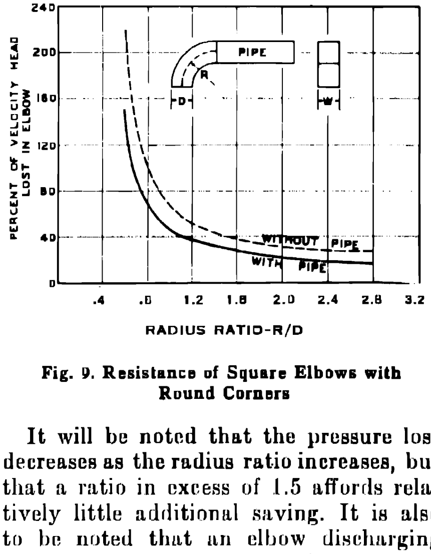

Fig. 9. Resistance of Square Elbows with Round Corners

It will be noted that the pressure loss decreases as the radius ratio increases, but that a ratio in excess of 1.5 affords relatively little additional saving. It is also to be noted that an elbow discharging directly has a greater loss than one followed by a section of straight pipe. In the latter case there is a regain of static pressure in the pipe as the crowded portion of the air stream in the outer portion of the elbow diffuses in the pipe beyond the elbow. The amount of the pipe regain varies from 30 to 40%.

A comparison of Figs. 9 and 10 shows that a rectangular elbow with a square outer corner is about 10% better than one with a rounded corner. Correction factors, by which to multiply the losses given in Figs. 9 and 10, when the duct is of rectangular cross section with various aspect ratios, are given in Fig. 11. **Aspect ratio** is the width of the elbow measured along the axis of the turn divided by the depth of the elbow measured along the radius of the turn. Fig. 11 illustrates an elbow with an aspect ratio greater than 1. It may be seen that the loss in a rectangular elbow decreases as the aspect ratio increases. This is just opposite to the variation in frictional resistance of straight pipe, where the loss increases if one dimension is much greater than the other.

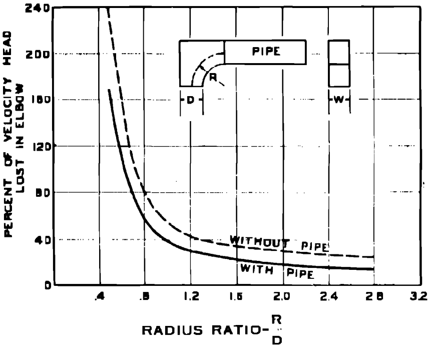

Fig. 10. Resistance of Square Elbows with Square Corners

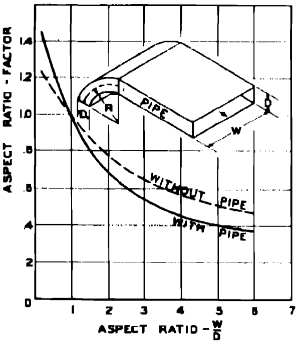

Fig. 11. Effect of Aspect Ratio on Resistance of Elbows

From these curves it may be deduced that right angle elbows may be used, with a very low pressure loss, if fitted with properly designed turning vanes, which really break up the elbow into a number of parallel elbows with advantageous radius and aspect ratios.

10. Pressure losses in changes of **cross sectional area** of the duct are given in **Fig. 12.** In Fig. 12 the change of section is abrupt. Curves A and B both give the loss in terms of the velocity head in the smaller pipe. In **Fig. 13** the change of section is gradual. The angle of divergence factor is applied to the per cent velocity head given by curve A of Fig. 12. For example, if the ratio A_1/A_2 is 0.5 (or $V_2/V_1 = 0.5$) the per cent velocity head lost if the change of section is abrupt is 25. However, if the change of section is gradual, with an angle of divergence, α, of 13°, the angle of divergence factor is 0.2. Hence the actual loss is

$$h_f = 0.2 \times 0.25\ h_v = 0.05\ h_v$$

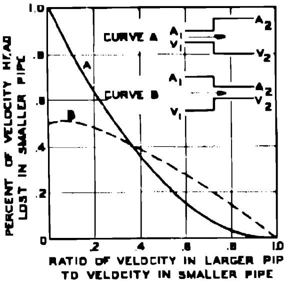

Fig. 12. Effect of Changes of Section

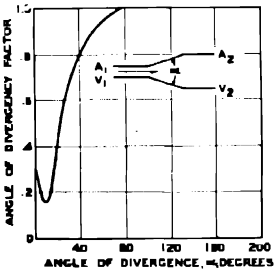

Fig. 13. Angle of Divergence, Degrees

A few special cases are given in **Fig. 15.** Fig. 15a is an elbow of less than 90°. Losses depend on the elbow angle, α. Multiply the losses for 90° elbows given in Figs. 8, 9, 10 and 11 by the elbow angle factor from **Fig. 14.**

Fig. 15b relates to the lost head when a duct discharges directly into a large space and Fig. 15c to the loss when the opening from a large space to a duct is flush with the wall of the space. Fig. 15d refers to the case where the duct projects somewhat into the space from which air is exhausted.

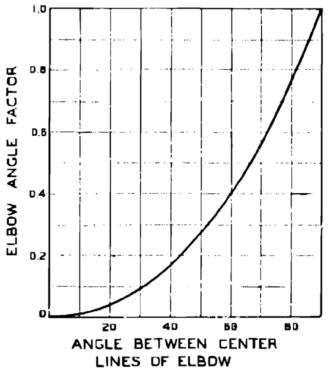

Fig. 14. Elbow Angle Factor

11. The simplest way to keep losses due to **duct turns in square elbows** at a minimum is to use large radius ratios and large aspect ratios. In case space limitations make this impracticable **duct turns with vanes** will reduce excessive loss, see **Fig. 16.** To determine the characteristics of vanes this formula may be used

$$N = \left(\frac{6D}{w} - 1\right)$$

where

D = depth of duct, in.

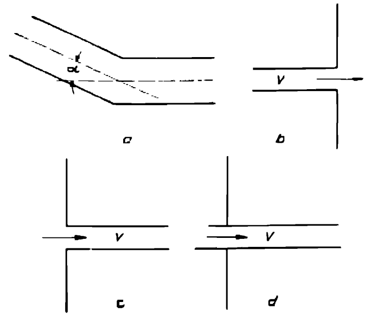

Fig. 15. Special Cases of Pipe Ends

a—Multiply values from Figs. 8, 9, 10 and 11 by angle factors from Fig. 13.
b—$h_f = 1.0 h_v$
c—$h = 0.48 h_v$
d—$h = 1.0 h_v$

w = width of duct, in.
N = number of blades (using nearest whole number)

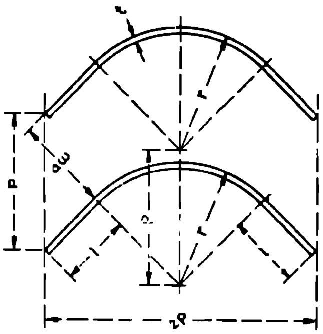

Fig. 16. Use of Duct Vanes

Again the axial width, Aw, has this relation

$$Aw = D/(N+1)$$

where

p = 1.14 Aw
r = 1.28 Aw
l = 0.75 Aw (lip on blade)
t = Aw/16 (thickness)

Blades should be as thin as possible without endangering loss of their shape.

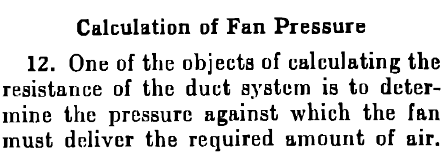

Fig. 17

Calculation of Fan Pressure

12. One of the objects of calculating the resistance of the duct system is to determine the pressure against which the fan must deliver the required amount of air. **Fig. 17** illustrates a fan exhausting air from and returning it to a room. The symbols used in Fig. 17 have the following significance:

V_d = Discharge velocity at the fan
h_{sd} = Static pressure at fan discharge

h_{fs} = Total resistance of the supply system, including the duct and heating and cooling surfaces (in blow-through systems)
V_s = Discharge velocity of air from the supply grilles
h_{ss} = Static pressure in the room
h_{fr} = Total resistance of the return system, including the duct, filters and heating and cooling surfaces (in draw-through systems)
V_i = Air velocity approaching the fan inlet
h_{si} = Static pressure at fan inlet

Writing Bernoulli's equation between points 1 and 2,

$$\frac{V_d^2}{2g}+h_{sd}=\frac{V_s^2}{2g}+h_{ss}+h_{fs} \tag{4}$$

and between points 3 and 4,

$$h_{ss}=\frac{V_i^2}{2g}+h_{si}+h_{fr} \tag{5}$$

Adding Equations (4) and (5), and transposing.

$$\left(\frac{V_d^2}{2g}+h_{sd}\right)-\left(\frac{V_i^2}{2g}+h_{si}\right)=\left(h_{fs}+h_{fr}+\frac{V_s^2}{2g}\right) \tag{6}$$

From the discussion on fans, it can be seen that the left side of the above equation is the total pressure of the fan. Hence

$$H_t=h_f+\frac{V_s^2}{2g} \tag{7}$$

where

H_t = Total pressure of the fan
h_f = Total resistance of supply and return ducts, filters and heating and cooling elements
V_s = Discharge velocity of air from the supply grilles

In the foregoing it is stated that the most generally used expression of fan static pressure is:

$$H_s=H_t-\frac{V_d^2}{2g} \tag{8}$$

where

H_s = fan static pressure

Substituting (8) and (7)

$$H_s=h_f+\frac{V_s^2}{2g}-\frac{V_d^2}{2g} \tag{9}$$

The result of this analysis is that, strictly speaking, the fan should be selected to have a static pressure equal to the sum of all the frictional and dynamic losses of the supply and return system plus the velocity head corresponding to the supply grille discharge velocity, but that the system should be credited with the velocity head corresponding to the fan outlet velocity. However, it is common practice to reserve the fan outlet velocity pressure as a factor of safety. This seems to be reasonable, especially since fan catalogs do not at present show the effect of the arrangement of the inlet connection to the fan on the fan pressure. It has been shown that this arrangement may have a very considerable influence on the fan pressure (4). Furthermore, transitions are frequently used between the fan outlet and the main trunk of the duct system which do not fully preserve the velocity pressure at the fan discharge. Therefore, it is common practice to select the fan on the basis of Equation (10). That is, the fan static pressure is calculated as that required to overcome the total resistance of the system plus the supply grille discharge velocity head.

$$H_s=h_f+\frac{V_s^2}{2g}. \tag{10}$$

Design Procedure

13. Any duct system consists of one or more trunks leading from or to a plenum chamber, fan or other common point. Regardless of which path the air follows, it finally issues from a grille into a room at atmospheric pressure. The pressure drop must, of necessity, be the same along each path, and hence the volumes of air flowing in each trunk or branch will automatically distribute themselves to fulfill this condition. It is the object of the ideal design to have this automatic distribution in accordance with the desired flow in each branch, and with all volume control dampers open.

The following logical procedure may be followed in designing a duct system:

a. Sketch a line diagram showing each

trunk and branch duct, and the location of each supply opening (make a separate similar sketch for the return-air system). Mark on the diagram the location of each elbow and change of section that will be required. This sketch should be based on a careful consideration of the building construction and of the most satisfactory arrangement of supply and return openings from the standpoint of air distribution. Care must also be exercised to keep the system as simple as possible.

b. Determine the air quantity from each supply opening on the basis of the **heat loss or heat gain** of the zones served by that opening.

c. Mark on each supply and return opening the **air velocity** to be employed, after having determined these from considerations of air distribution.

d. Mark on each section of the system the **amount of air** flowing in that section.

e. Determine the **sizes** of all main and branch ducts by first assigning a tentative velocity for each section and then calculating the duct size from the air volume and velocity. Trunk velocities should be reduced as the distance from the fan increases, and short branches near the fan should have higher velocities than long branches far away.

f. Calculate the **pressure drop** through several of the more important paths. If these differ from each other by more than approximately 20%, increase the velocities in the paths with the lower resistances or decrease the velocities in the paths with the higher resistances. Finally recalculate the pressure drops.

g. Estimate whether all remaining paths have approximately the same pressure drop. If in doubt, calculate the pressure drops.

h. Repeat the above procedure for the return air system.

i. The total duct resistance is the sum of the resistances of a supply and return system.

Another method that is frequently employed, after completing steps a to d above, is to determine the sizes of all main and branch ducts on the basis that the friction loss per foot will be the same in all ducts. Unless the system balance is to be entirely dependent on damper settings, this method offers the disadvantage that the resistances of the shorter branches are automatically lower than those of the longer branches. However, it may be used as a starting point for step e in the method outlined above, especially when the pressure available at the fan is specified.

14. The following **example,** based on the simple duct system for a small cooling job in **Fig. 18,** illustrates the application of the data and procedure described above. This example is based on the premise that the duct system must be designed to have a pressure drop not to exceed 0.30 in. H_2O.

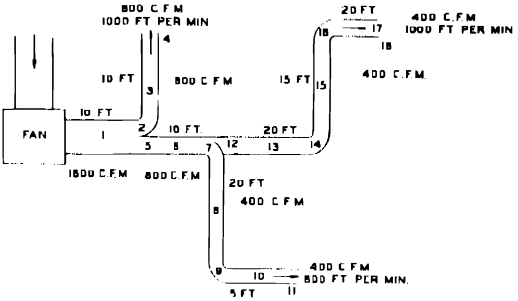

Fig. 18. Example of Pressure against Fan

The quantity and velocity of air discharged from each grille is determined to meet the requirements for cooling and air distribution.

The pressure drops through grilles are determined from catalog data, and the discharge velocity head is calculated from Equation (3). The sum of these two may be termed the total discharge loss. These may be tabulated as follows:

Grille No.	*4*	*11*	*18*
Grille pressure drop	0.050	0.032	0.050
Discharge velocity head	0.062	0.040	0.062
Total discharge loss	0.112	0.072	0.112

As a first approximation, assume that one-third of the duct losses occur in the return system. This leaves 0.20 in. for the

Table 2. Duct Design Example

Duct section	Round diam, in.	Rectangular dimensions, in.	Velocity, fpm	Friction per 100 ft, in. H_2O	Total friction, in. H_2O	Velocity head, in. H_2O
1	15	8 ×25	1150	0.18	0.018	0.083
3	11	8 ×13	1110	0.20	0.020	0.077
6	11	8 ×13	1110	0.19	0.019	0.077
8, 10	9	8 × 9	800	0.15	0.037	0.040
13, 15, 17	12	8 ×15	480	0.035	0.019	0.014

supply system, Fig. 18. Design the longest path (to grille No. 18) to have a pressure loss not to exceed 0.20 in., and then balance the other two paths for the same loss. The total discharge loss is 0.112, leaving 0.088 in. for friction, elbows, etc. Assume that one-third of this is dynamic loss. This leaves 0.058 in. for friction alone. The total length of the run is 75 ft. Hence the allowable friction loss is 0.077 in. per 100 ft if all sections were to have equal friction per ft. However, in order to help balance the pressure drops in the paths to grilles 4 and 11 it is advisable to increase the velocities slightly in sections 1 and 6, and to decrease the velocity in section 13. Assume that the losses in sections 1, 6 and 13, 15, 17 are each one-third of the allowable 0.058. Determine the allowable friction loss per 100 ft and from Fig. 6 select the correct size of round duct.

Section No.	*1*	*6*	*13*
Allowable friction	0.019	0.019	0.019
Allowable friction per 100 ft	0.19	0.19	0.035
Cfm	1600	800	400
Round duct diameter, in.	15	11	12
Actual friction per 100 ft	0.18	0.19	0.035
Velocity in round duct, fpm	1300	1210	510

Table 3

Section No.	Dynamic loss, in. H_2O	Nature of loss
2	0.021	Elbow
5	0	Velocity reduction
7	0.013	Elbow
9	0.010	Elbow
12	0.013	Velocity reduction
14	0.004	Elbow
16	0.004	Elbow

Section 3 must have a friction equal to twice that of section 1, or about 0.36 in. per 100 ft. However, in Fig. 6 it is found that this corresponds to a velocity of 1,400 fpm. This would introduce a considerable dynamic loss in section 2. Hence, compromise with an 11-in. round duct, having a velocity of 1,200 fpm and a friction loss of 0.20 in. per 100 ft.

The allowable loss for sections 8 and 10 is 0.019 plus the difference between the discharge losses at 18 and 11, making a total of 0.059, or 0.24 in. per 100 ft. Use a 9-in. round duct, having a velocity of 910 fpm and a loss of 0.15 in. per 100 ft.

Note that in the above preliminary layout it is not necessary to go to the extreme of using fractional diameters in the round duct sizes.

Now select from Fig. 7 the dimensions of rectangular ducts corresponding to the round sizes calculated above. Calculate the velocity and velocity head in each duct section. It is assumed here that a duct having a depth of 8 in. is convenient. See **Table 2.**

Dynamic losses will now be calculated at the various sections. All elbows will be assumed to have a radius ratio of 1.5. From Fig. 9, the per cent of velocity head lost in a square elbow is 26.

The take-off at section 2 is considered an elbow with aspect ratio $8/13 = 0.62$. From Fig. 11 the aspect ratio factor is 1.2. Hence the loss at 2 is

$$1.2 \times 0.26 \times 0.066 = 0.021 \text{ in.}$$

At section 5 there is no change in velocity so that the entry loss is negligible.

In the take-off at 7 there are losses from the elbow effect and from the velocity decrease. However, the latter is small (velocity ratio = 0.72). The aspect ratio of

Table 4. Summary of Example

Grille No. 4		Grille No. 11		Grille No. 18	
Duct section	Friction	Duct section	Friction	Duct section	Friction
1	0.018	1	0.018	1	0.018
2	0.026	5	0	5	0
3	0.020	6	0.019	6	0.020
4	0.112	7	0.020	12	0.022
		8, 10	0.040	13, 15, 17	0.014
		9	0.010	14	0.003
		11	0.072	16	0.003
				18	0.112
Total	0.176		0.179		0.191

this elbow is 0.89, and from Fig. 11 the aspect ratio factor may be taken as 1.0. Assume the loss in this elbow to be 26% of the velocity head in section 6.

The dynamic losses at sections 9, 12, 14 and 16 may be calculated in a similar manner. These are summarized as in Table 3. The total pressure drop in each path may now be calculated as shown in Table 4.

Of course these calculations are not perfectly accurate. Adjustable **splitter dampers** may be installed at sections 2 and 7. If this is not convenient, volume dampers may be installed in each branch.

Table 5. Typical Single Inlet Centrifugal Blower Wheels (Torrington)

Dimensions		No. of Blades	Sp	Cfm	Hp	Sp	Cfm	Hp
Diam	Width							
				1,725 rpm			3,450 rpm	
$4\frac{1}{4}$	$2\frac{1}{8}$	30	0	220	0.035	0	440	0.28
			0.2	190	0.030	0.8	380	0.24
			0.4	150	0.025	1.6	300	0.20
				1,140 rpm			1,725 rpm	
$5\frac{1}{4}$	$3\frac{7}{16}$	32	0	230	0.021	0	350	0.075
			0.1	195	0.018	0.3	280	0.056
			0.2	155	0.015	0.6	156	0.030
$6\frac{5}{16}$	$3\frac{1}{8}$	32	0	360	0.050	0	550	0.167
			0.2	285	0.042	0.5	430	0.130
			0.4	180	0.037	1.0	220	0.075
$7\frac{27}{32}$	$4\frac{15}{32}$	36	0	750	0.160	0	1,160	0.62
			0.2	670	0.140	0.5	990	0.52
			0.4	560	0.120	1.0	840	0.44
$9\frac{11}{32}$	$5\frac{5}{16}$	36	0	1,160	0.35	0	1,750	1.20
			0.3	1,050	0.30	1.0	1,490	1.00
			0.6	900	0.26	2.0	1,060	0.65
				860 rpm			1,140 rpm	
11	$5\frac{1}{4}$	42	0	1,225	0.033	0	1,620	0.075
			0.2	1,140	0.029	0.6	1,420	0.06
			0.4	1,030	0.026	1.2	1,060	0.043

References

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2. Heating Ventilating Air Conditioning Guide, Am. Soc. of Heating and Ventilating Engineers, New York, 1950, Chap. 31.
3. Wirt, Loring, New Data for the Design of Elbows in Duct Systems, *G. E. Review*, p. 286, June 1927.
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37. AIR CONDITIONING CALCULATIONS

AIR conditioning is defined as the simultaneous control of temperature, humidity, air motion and air purity in any given space. This definition gives rise to the terms "winter air conditioning," "summer air conditioning," "complete air conditioning," when considering air conditioning for comfort. Industrial air conditioning is not considered in this chapter.

Winter air conditioning includes heating, humidification, air distribution and air cleaning, when outside temperatures are below the inside or room temperature. It is the well known science of heating and ventilating, involving the calculation of heat loss from conduction through building construction, heating requirements for ventilating air, and heating requirements for humidification.

Summer air conditioning includes cooling, dehumidifying, air distribution and air cleaning, when outside temperatures are above the inside or room temperature. It involves the calculation of heat gain from conduction through building construction, direct sunlight heat gain through glass and indirect sunlight heat gain through opaque building construction, heat gain from lights and motors (including fan motor) within the conditioned space; as well as heat and moisture gain from people, from outside air required for ventilation, and from appliances and other heat and moisture generating equipment within the conditioned space.

Complete air conditioning includes both heating and cooling, and both humidifying and dehumidifying, together with air distribution and air cleaning. It therefore involves all the calculations for both winter and summer air conditioning.

Design Conditions

1. The first step in the approach of an air conditioning problem is to determine the proper design conditions. Outside conditions are based on U.S. Weather Bureau statistics, and inside conditions upon the type and duration of occupancy, and also upon the outside temperature.

Inside conditions for cooling. The determining factor for inside design conditions is comfort measured by the chart shown in **Fig. 1** in terms of **effective temperature,** which is an index of conditions producing the same degree of comfort (see Chap. 35). Design conditions have also been recommended for various cities in the United States as in Table 2, Chap. 11. Inside conditions are given in terms of effective temperature, separately for occupancies under 40 minutes and for occupancies over 40 minutes. Inside temperature and humidity conditions corresponding to the recommended effective temperature can be determined by reference to Fig. 1. For inside conditions for cities not listed in Table 2, Chap. 11, reference should be made to the comfort chart of Fig. 1. An effective temperature condition within the comfort zone should be chosen for cooling such that the differential between the inside and outside dry bulb temperature is as low as practicable. Temperature differentials of from 10 to 20 deg are most desirable, and differentials of 25 to 30 deg should be avoided if possible. However, inside conditions higher than 80 to 82 F dry bulb are seldom very comfortable in practice, as the very low humidities required for comfort at high dry bulb temperatures are difficult to maintain. Therefore, in very warm climates, a differential of 25 to 30 deg may be necessary at extreme outside conditions. Normally, humidities are kept within the range of 35 to 65%.

Inside conditions for heating. For heating, the recommended practice for inside design conditions for human comfort are 70 F dry bulb temperature and 35% relative humidity with 30 F outside temperature. In warmer climates where the outside temperature seldom gets down to freezing, it is often desirable to maintain inside temperatures from 72 to 75 F.

Outside conditions. The tables of Chapter 11 give recommended outside design conditions for both heating and cooling seasons. For cities not listed in this table, reference can be made to local Weather Bureau reports for maximum and average maximum conditions of dry bulb and wet bulb temperatures for the warmest month for cooling,[1] and to the minimum and average minimum dry bulb temperatures for the coldest month for heating.[2] The **outside design conditions** should be somewhere between the maximum and average maximum for cooling, and between the minimum and average minimum for heating. It is seldom good engineering to design for the extreme maximum or absolute minimum outside conditions.

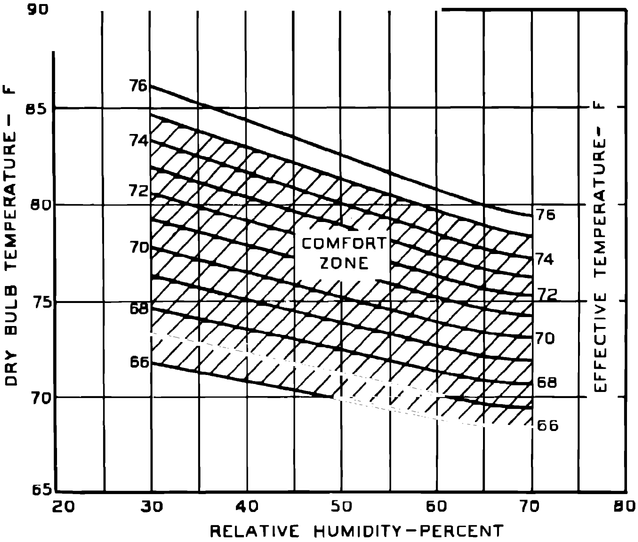

Fig. 1. Summer Comfort Zone Section of Comfort Chart for Air Motion 15-25 fpm Using Basic Effective Temperature

Often wet bulb temperatures are not obtainable from Weather Bureau data, but relative humidities are usually given. Wet bulb temperatures may be obtained by taking simultaneous conditions of dry bulb and relative humidity and referring to the psychrometric chart as in **Fig. 2.**

In the absence of more specific data for a required locality not listed in tables of Chap. 11, compare its Weather Bureau statistics with similar statistics for listed cities to determine which is nearest climatically. Then the design conditions (both inside and outside) may be selected from this table for a city having a similar climate to the required locality. However, the comparison may require using one city's heating design conditions and another's cooling design conditions to approximate closely the climate of the required locality.

Sensible Heat Load

2. After selection of the proper design conditions, the first step in the air conditioning calculations is to determine the sensible head load. **Heat leakage calculations**[3] for summer air conditioning are similar to heat loss calculations for heating except that the flow of heat is reversed. In the heat gain calculations, the phenomenon of **radiant heat** gain from the sun plays a prominent part. Furthermore, heavy building construction, particularly masonry walls, has a considerable heat absorbing capacity which interposes a lag in the heat flow which often must be considered for reasonably accurate results in heat gain calculations.

It is often desirable in air conditioning computations to estimate the temperature in **attic spaces** between the roof and the furred ceiling below. This can readily be done as the heat transfer from the outside to the attic space must equal the heat transfer from the attic to the conditioned space below. The following formula may be used, solving for X, the attic temperature:

$$SU(T-X) = su(X-t)$$

Where

S = roof area in sq ft
s = ceiling area in sq ft
U = roof transmission factor, Btu per sq ft hr F

Table 1. Direct Sunlight Gain on Glass[4,5]

(Btu per hr sq ft net glass area)*

Latitude	20°			30°			40°		
	July	Aug.	Sept.	July	Aug.	Sept.	July	Aug.	Sept.
Northeast	148	126	91	144	118	94	138	109	58
East	178	181	179	179	181	174	179	178	165
Southeast	86	120	149	104	137	165	132	157	176
South	—	14	60	19	58	107	65	104	148
Southwest	86	120	149	104	137	165	132	157	176
West	178	181	179	179	181	174	179	178	165
Northwest	148	126	91	144	118	94	138	109	58
Flat skylight	263	258	242	256	244	218	242	220	185

* Values given represent the maximum for each exposure.

For light-colored shades or inside venetian blinds, use 60% of unshaded values if shade is at least ⅔ drawn. For outside ventilated awnings and venetian blinds arranged to cut off all direct sunlight use 15 Btu per sq ft per hr for all exposures instead of values in table; for ordinary awnings not ventilated, use 25 Btu per sq ft per hr. When using sash area normally having about 85% glass area, multiply above values by 0.85. Use other appropriate correction for different percentage of glass.

u = ceiling transmission factor, Btu per sq ft hr F
T = outside roof temperature, F
t = inside conditioned space temperature, F
X = attic temperature, F

To consider carefully all of the complexities of variable heat flow through building construction would make the calculations unduly long and tedious. It is, therefore, common practice in commercial work to make a number of approximations which simplify the problem and in the total give reasonably accurate results. For shaded outside walls or roofs, the full temperature difference between the inside and the outside design conditions is used in the heat gain calculations, except in the case of extremely heavy masonry walls in climates where the nights are considerably cooler than the days.

Sun effect. For sunlit walls it is safe to use an equivalent outside temperature of 120 F in the United States, and for sunlit roofs an equivalent outside temperature of 140 F. For sunlit glass[4,5] add radiant heat gain in accordance with values given in **Table 1**. This radiant heat gain on sunlit glass is in addition to the transmission gain through the glass figured in the usual manner from the transmission factor, glass area, and the temperature differential. When calculating total cooling load, radiant heat gain from the sun need only be figured on the roof and on the exposure having the greatest radiant heat load.

When roofs or walls are painted with aluminum or have highly reflective, light-colored surfaces, the radiant heat gain through such walls or roofs may be reduced 50%. In other words, the surface temperature for roofs may be reduced to 120 F and walls to 110 F, when using an outside design condition of 100 F.

Heat lag. The matter of lag through heavy building construction is much harder to account for in the heat gain calculations. In many cases, the peak of wall transmission may not coincide with peak load for the cooling system due to this time lag in the transmission of heat. Ordinary frame construction such as used in residential work has but little capacity effect and full temperature differences should be used. Masonry wall time lag varies from 3 hr on a 6-in. concrete wall to 10 hr on a 22-in. thick, solid brick and tile, composition wall. Thus the heat from the sun effect on an east exposure may just be getting through the wall when the sun has moved around and is starting to shine on the west exposure.

The effect of this lag is usually averaged by figuring sun effect only on one exposure, the one having the greatest area, and the roof. For example, if the south wall has the greatest sunlit area of glass and wall surface, the sun effect should be figured on the south exposure and not on the east and west. When extremely heavy masonry west walls are encountered, the radiant heat may not get through these walls until after the building occupants go home for the evening. In such cases, no radiant heat gain need be figured for the west wall, but should be figured for either east or south exposures, if they are sunlit.

per hr. Part of this is sensible heat, and the remainder is latent heat which adds moisture or humidity to the air. The total heat from people increases enormously as their activity increases, this increase being largely due to the greatly increased latent heat produced. Fig. 3 gives both sensible and latent heat quantities produced by people at rest, and at various conditions of activity, for different room temperatures.

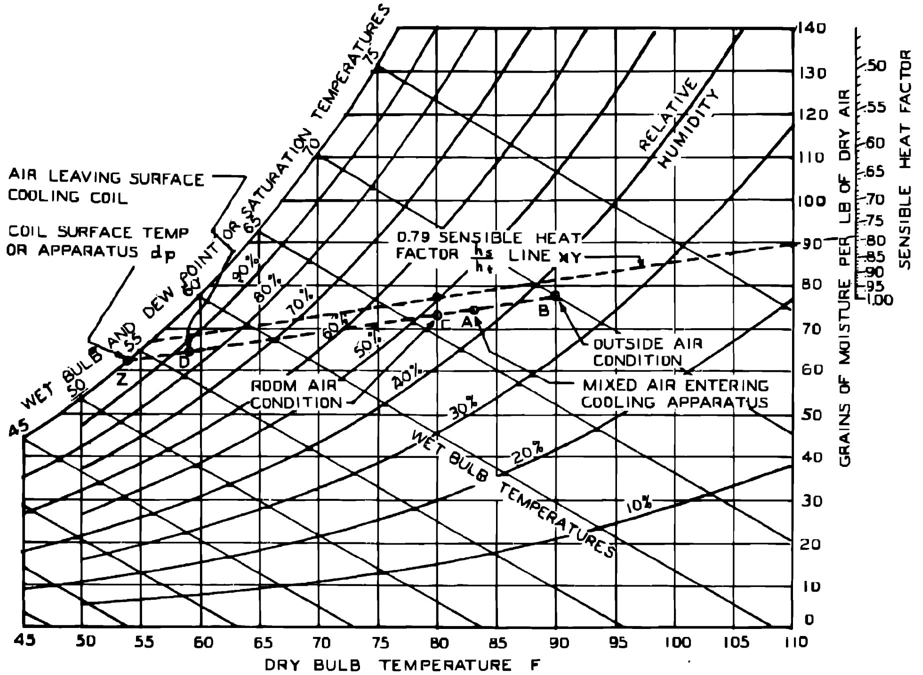

Fig. 2. Psychrometric Chart (See Example II)

Internal heat. In addition to the transmission and radiant heat gain through walls, floors, ceilings and glass areas, there must be added to the sensible heat load, the heat gain from any objects in the conditioned space that give off heat,[6] and the heat gain from the outside air required for ventilation or to offset infiltration. Under this heading come people, lights, motors (including heat equivalent of brake horsepower of fan motor used for air circulation), and any heat evolved in processing.

People. The total heat evolved by people at rest amounts to approximately 400 Btu

Food. For restaurants an allowance of about 30 Btu per person should be added, to account for the sensible heat gain from the food. However, where the heat generating equipment such as steam tables, coffee urns, etc., is in the conditioned room and is accounted for in the heat gain calculations (as in the case of a cafeteria), this allowance of 30 Btu per person is not included in addition.

Electricity. The heat generated by lights, motors, and electrical equipment within the conditioned spaces must be included in the sensible heat load. Lighting or electrical equipment load in watts multiplied by the factor 3.4 will give the heat equivalent in Btu per hr. Often the electrical equipment will not be operating continuously and the proper time factor should be applied in such cases. **Table 2** gives approximate values for heat gain from motors located within the conditioned space. Usually motors are not fully loaded besides not operating continuously. Therefore, the proper load and operating-time factors should be applied to reduce the heat gain accordingly. An average overall factor for a considerable number of motors operating in the conditioned spaces is 50%.

Table 2. Heat Gain from Motors

Nameplate rating, hp	Heat gain in Btu per hr per hp	
	Connected load in same room	Connected load outside of room
$\frac{1}{8}$ to $\frac{1}{2}$	4,250	1,700
$\frac{1}{2}$ to 3	3,700	1,150
3 to 20	2,950	400

The heat gain from the fan motor used to circulate the conditioned air must also be included in the sensible heat calculations. Only the heat equivalent of the fan brake horsepower need be added since the motor heat losses will not be absorbed in the air stream. It is common practice, however, to use the nameplate horsepower (rather than the actual brake horsepower) for an additional factor of safety, multiplying the horsepower by 2,545 to obtain the heat equivalent in Btu per hr.

Outside air. To the sensible heat gains, calculated as explained above, must be added the sensible heat gain from infiltration and from outside air introduced, to obtain the total sensible heat gain. Infiltration may be figured by the Application Engineering Standards for Air Conditioning for Comfort of the ACRMA.[4] The latter gives factors for infiltration through swinging and revolving doors, for infiltration through cracks around windows and doors, and for ventilation standards (outside air requirements) for various applications.

However, for simplicity, an arbitrary method of offsetting infiltration when cooling, by figuring a minimum of a one hour air change of outside air in the conditioned space, will give reasonable approximate results. Thus, when the outside air introduced is greater than the equivalent of a one-hour air change, no infiltration need be figured. If the actual outside air requirement is less than a one-hour air change, an additional amount should be added for infiltration so that the total will be equivalent to a one-hour air change. This minimum outside air quantity is in addition to any exhaust that may be taken from the conditioned space.

Table 3. Heat Gain from Restaurant Equipment*[6]

	Sensible heat in Btu per hr	Latent heat in Btu per hr
Coffee urns (steam) per gal capacity	800	800
Coffee urns (electric) per gal capacity	700	500
Coffee urns (gas burning) per gal capacity	800	800
Steam tables (per sq ft top surface)	400	800
Steam tables (electric) per sq ft top surface	200	350
Steam tables (gas burning) per sq ft top surface	850	430
Gas toasters (640 slices per hr)	12,000	5,000
Gas toasters (360 slices per hr)	7,700	3,300
Gas toasters (small size)	5,750	830

* When equipment is hooded with positive exhaust above values may be reduced 50%.

Latent Heat Load

3. The second step in the cooling load calculations is to determine the latent heat load. This load comes from the moisture pickup or humidity rise in the conditioned space, and from the excess moisture in the outside air if the outside dew point is higher than the inside design dew point. The moisture pickup in the conditioned space comes from people and any special processes that give off moisture.[6] **Fig. 3** gives the latent heat quantity produced by people at rest and at various conditions of activity.

In dealing with cafeterias and restau-

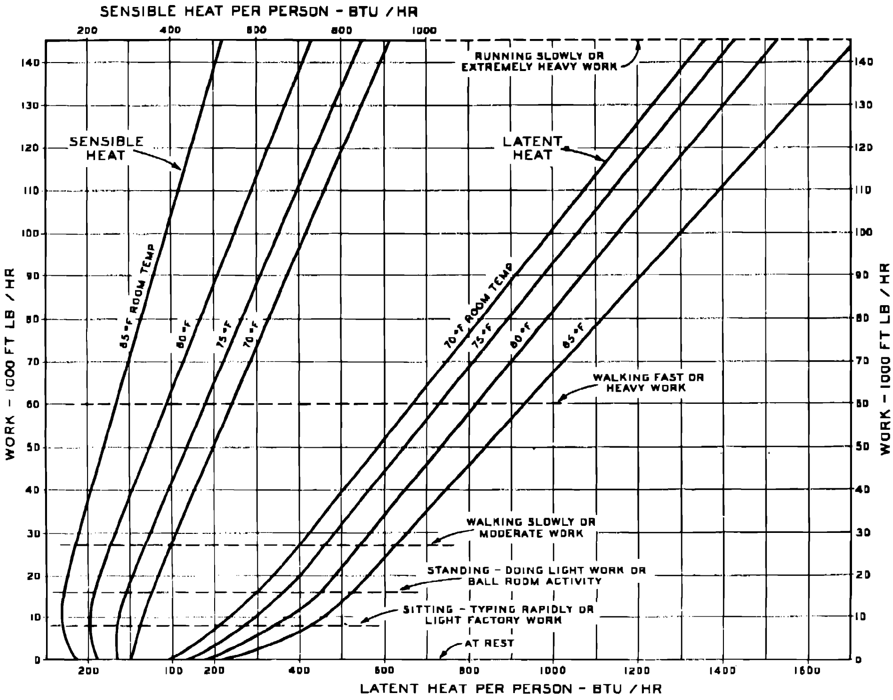

Fig. 3. Heat Gain from People, Btu per hr

rants we have many sources of moisture load to consider. Table 3 gives both sensible and latent heat loads for coffee urns, steam tables, and other restaurant equipment. If such equipment is adequately hooded with a positive exhaust, both the sensible and latent heat loads may be reduced 50%. If the steam tables, coffee urns, etc., are in the conditioned space, no additional load is ordinarily figured for the food. Otherwise a factor of 30 Btu per person should be added to account for the latent heat from the food.

The heat equivalent of moisture removed is 1,060 Btu per lb and there are 7,000 grains of moisture per lb. Thus to change grains of moisture removed to latent heat load in Btu divide by 7,000 and multiply by 1,060.

Air Quantities

4. Outside air. The outside air quantity must be determined before the total sensible and total latent heat loads can be calculated. As previously explained, the minimum outside air quantity for use in determining heat loads is equivalent to a one hour change of air in the conditioned space. For ventilation requirements[4] it is good practice to figure 10 cfm of outside air per person—25 to 40 cfm if they are smokers. In crowded places such as theaters, where the concentration of people is very great, an absolute minimum of 5 to $7\frac{1}{2}$ cfm of outside air is often figured for peak load conditions.

Supply air. The total air quantity circulated is a function of the internal sensible heat load and the temperature difference between the supply air and the conditioned room. The supply air temperature for cooling usually varies from 10 to 30 F below the room temperature. With the advance in air distributing methods, and in grille and diffuser design, temperature differences of 20 to 30 F are being increasingly used without causing uncomfortable drafts.

The immediate method of calculating the supply air quantity is to divide the total internal sensible heat load (outside air and fan motor loads not included) by the heat taken up per pound of air circulated. Thus, the sensible load in Btu per min, divided by the temperature difference (between the supply air and the room) and the specific heat of air (0.24), will give the pounds of air circulated per minute. Multiplying this value by the specific volume will give cubic feet of air per minute.

Return air. The return air quantity is, of course, merely the difference between the supply and the outside air quantities. Many engineers prefer to use pounds of air per minute in all the calculations, rather than cubic feet per minute, to avoid confusion due to the different specific volumes at various points in the air cycle. If the calculations are carried through using cubic feet per minute it simplifies the computations, and it is common practice to figure all air quantities in cubic feet per minute based on the specific volume at room conditions.

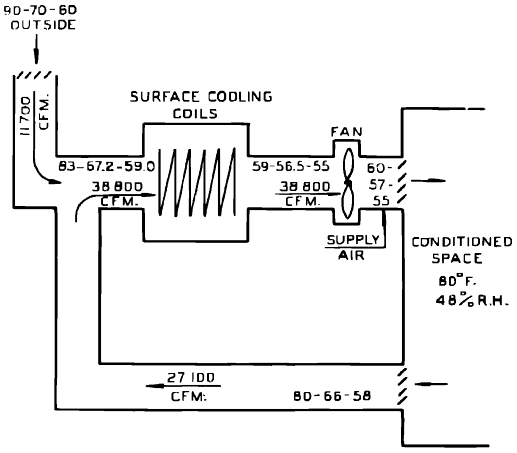

Fig. 4. Surface Cooling Cycle, Showing Dry Bulb, Wet Bulb and Dew Point Temperatures at Various Points (Example II)

Load Calculations

5. The most important step in the refrigeration load calculations is the preparation of the "Btu sheet," the heat load estimate. **Example I** is a typical calculation sheet for a medium size department store. At the top, the various air properties corresponding to the outside and inside design conditions should first be set down. The outside dry and wet bulb design conditions are obtained from Table 2, Chap. 11, or other weather data available as previously described. The inside design conditions are determined by reference to Figs. 1 and 2, and by considering the phenomenon of effective temperature. The various other air properties necessary for the air conditioning calculations can then be determined from the psychrometric chart, and tables of air properties (see Chapters 4 and 10).

In Example I heat lag is accounted for by figuring all exposures shaded excepting the roof and the south wall. On shaded exposures the full temperature difference is used in the calculations although additional lag may be present. No additional factor of safety need be added, when calculating in this manner, except in unusual cases as the factors used are sufficiently safe to cover reasonable duct and apparatus losses.

The "dry tons" (sensible load) and "moisture tons" (latent load) should be calculated separately, as indicated in Example I. Having calculated these quantities, the next step is to determine the sensible-total heat ratio h_s/h_t, which is merely the total dry tons divided by the total refrigeration load (the sum of the dry tons and the moisture tons.)

Equipment Selection[7]

6. Surface cooling.[8] Before the cooling equipment is selected to meet the calculated load requirements, the supply air, outside air and return air quantities must be decided upon in accordance with methods already described. In selecting surface cooling coils, the governing factors are the entering air dry bulb and wet bulb temperatures (which determine the relative humidity), and the evaporator temperature. Having arrived at the air quantities, it is merely a matter of arithmetic to determine the entering air conditions by ref-

Example I. Human Comfort Air Conditioning Calculations

	Winter	Summer		E.T.	rh	dp	gr/lb	Btu/lb	cu ft/lb
	db	db	wb						
Outside	35	90	70	—	37%	60	77.29	34.04	14.12
Inside	70	80	66	73.5	48%	58	71.86	30.79	13.84

Department store, Los Angeles, California. Type of room—First floor sales area with lunch counter
Room volume (Length 169½ ft × width 103½ ft × height 15½ ft = 272,000 cu ft

Sensible Load (Dry Tons)	Cooling				Heating		
Heat Leakage	Net sq ft of surface	Heat transfer factor U	Temp diff F	Btu per hr	Heat transfer factor U	Temp diff F	Btu per hr
Walls or portions of walls exposed to outside							
North	—	—	—	—	—	—	—
South-shaded	—	—	—	—	—	—	—
South-unshaded	950	0.30	40	11.400	0.30	35	9,975
East	1,300	0.30	10	3.900	0.30	35	13.650
West-shaded	570	0.30	10	1.710	0.30	35	5.985
West-unshaded	—	—	—	—	—	—	—
Walls or portions of walls directly in contact with ground	—	—	—	—	—	—	—
Glass in outside walls							
North	—	—	—	—	—	—	—
South-shaded (entrance doors)	240	1.13	10	2,712	1.13	35	9,500
South-unshaded (show windows and backing)	1,435	0.45	40	25,830	0.45	35	22,600
East	305	1.13	10	3,447	1.13	35	12,100
West-shaded (entrance doors)	120	1.13	10	1,356	1.13	35	4,750
West-shaded (show windows and backing)	915	0.45	10	4,118	0.45	35	14,400
Interior walls or portions of walls next to unconditioned areas							
North	2,625	0.25	5	3,281	0.25	0	—
Ceilings or portions of ceilings							
Under flat roof	17,550	0.10	60	105,300	0.10	35	61,425
Under attic	—	—	—	—	—	—	—
Under fin. rooms	—	—	—	—	—	—	—
Skylights	—	—	—	—	—	—	—
Floor or portions of floor							
Over fin. rooms	—	—	—	—	—	—	—
Over basement	17,550	0.35	5	30,713	0.35	25	153,565
Over ground	—	—	—	—	—	—	—
Sub-total				193,767			307,950

	Cooling				Heating	
Internal Heat	Unit	No. of units	Factor	Btu per hr		Btu per hr
Outside air — As infiltration and/or air through conditioner	cfm	11,700	$\frac{(0.24)(60)(10)}{13.84}$	121,660	$\frac{(0.24)(60)(35)}{13.47}$	437,580
People (normal no. of people)	person	1,170	220	257,400	Add to heating duty for latent load:	
Lights (watts on sunny day)	watts	87,750	3.4	298,350		
Other sources of heat					Heat of vaporization for humidifying = lb water per hr × 1,060 = None × 1,060 = 0	
1. Coffee urn—steam (hooded)	gal	3	(.50) (800)	1,200		
2. Steam tables electric (hooded)	sq ft	4	(.50) (200)	400		
3. Motors—supply fan	hp	15	2,545	38,175		
4. Motors—three ⅓ hp = 1 hp total	watts	1,250	3.4	4,250		
5. Electrical apparatus (hooded)	watts	27,650	(.50) (3.4)	47,000		
6. Gas apparatus (hooded)	small toaster	2	(.50) (5,750)	5,750		
		Total sensible cooling load = 967,952			Gross heating load	745,530
		Total sensible load, tons $= \frac{967,952}{12,000} = 80.7$			Minus lights	298,350
					Net heating load	447,180

Example I. Human Comfort Air Conditioning Calculations. (Continued)

Latent load (Moisture tons)	Cooling				Cooling load summation
Internal Heat	Unit	No. of Units	Factor	Btu per hr	
Outside air As infiltration and/or air through conditioner	cfm	11,700	$\frac{(1{,}060)\ (60)\ (5.43\ \text{Gr})}{(13.84)\ (7{,}000)}$	41,650	Total sensible heat load, tons 80.7
People	person	1,170	180	210,600	Total latent heat load, tons 21.2
Other sources of moisture					Total load, tons 101.9
1. Coffee urns—steam (hooded)	gal	3	(.50) (800)	1,200	Sensible-total heat ratio $= \frac{h_s}{h_t}$
2. Steam tables—electric (hooded)	sq ft	4	(.50) (350)	1,700	
3. Gas apparatus (hooded)	small toaster	2	(.50) (830)	830	$= \frac{80.7}{101.9} = 0.79$
	Total latent heat load, Btu per hr			254,980	
	Total tons latent heat load $= \frac{254{,}980}{12{,}000} = 21.2$				

erence to the tables of air properties. Surface cooling coil tonnage ratings are found from manufacturers' tables at various entering air conditions, evaporator temperatures and sensible-total heat ratios, h_s/h_t. It is usually necessary to refer to the psychrometric chart, Fig. 2, at some time during the process of selection. It is also often necessary to vary the air quantities (particularly the supply air quantity), in order to obtain an economical coil selection. **Example II** illustrates typical calculations for the selection of surface cooling coils for the cooling load calculated in Example I. (Conclusions based on a coil manufacturer's data.)

Air washer cooling.[9] When using an air washer for dehumidifying, the governing factors in the selection are the entering air wet bulb temperature, and the required leaving air dew point temperature. The supply, outside and return air quantities are determined in the same general manner as for surface cooling, but in addition the dehumidified air quantity must be calculated. Only sufficient air is dehumidified, by going through the air washer, to balance the total sensible load. The dehumidifier leaving air dew point is determined from the latent load requirements.

For most comfort cooling applications, the dehumidified air quantity will be the supply air quantity, as it is unusual for the dehumidifier leaving air temperature to be more than 30 deg below the required room temperature. Where the supply air quantity, or "room diffusion" (difference between room and supply air temperature), is definitely fixed by specifications, or in cases of unusual air distribution problems requiring a low room diffusion, it may be necessary to bypass a portion of the supply air around the air washer. Most air washers are provided with a bypass having automatic dampers for control, but the continuous bypassing of a minimum amount (as in **Example III**) is not ordinarily required. However, Example III has been figured with the same air quantities as Example II for purposes of comparison between surface cooling and dehumidifier calculations.

As in the case of surface cooling, the psychrometric chart, and the sensible-total heat ratio, h_s/h_t, are used to determine the dehumidifier leaving air temperature (or dew point in a saturating type washer, as is generally used for comfort air conditioning). Example III illustrates the calculations for the selection of an air washer for the cooling load calculated in Example 1. The dehumidifier leaving air temperature, as obtained by using the sensible-total heat ratio on the psychrometric chart, should always be checked arithmetically to see that it is sufficiently low to balance the calculated latent heat load. Using moisture values from the tables at the entering and leaving air dew points, the actual moisture removal and latent heat equivalent thereof may be calculated as indicated in Example III. If the moisture removal is not enough to balance the latent heat load, a lower leaving air dew point will be required and the dehumidified air quantity adjusted accordingly. It is also necessary, when using an air washer, to increase the total refrigeration lead by the heat equivalent of the horsepower of the pump used to circulate the spray water.

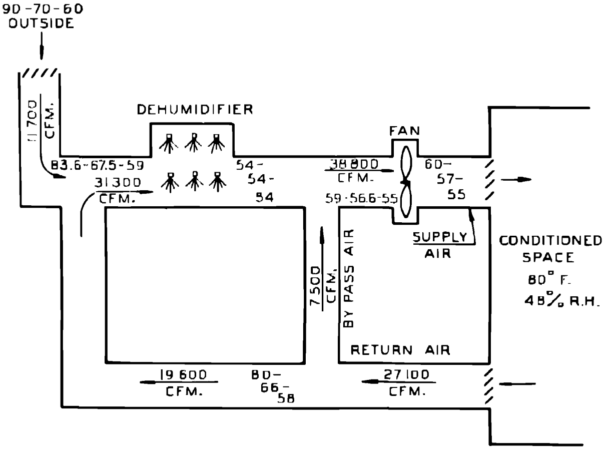

Fig. 5. Dehumidifier Cycle, Showing Dry Bulb, Wet Bulb and Dew Point Temperatures at Various Points (Example III)

Conclusion

7. Cooling load calculations are very similar to heating calculations, but are considerably more complicated in that the air conditioning engineer must be thoroughly familiar with the following special circumstances which affect the design[10]:

1. Inside design conditions for comfort
2. Outside design conditions
3. Sunlight heat gain and heat lag
4. Ventilation requirements
5. Air properties and the psychrometric chart

On a large building[11] where the solar radiant heat gain forms a large percentage of the cooling load, the air distributing system should be divided into sections (called zones) so that the cooling may be varied in accordance with the movement of the sun effect across the building. Preferably, each exposure should be zoned separately, but north and east exposures may be put on one zone, and south and west exposures on another. The radiant heat load should be figured for each zone to determine the maximum air requirements for the zone. However, when calculating the total refrigeration load for the building, only the radiant load on the roof and one exposure need be included as the maximum solar heat gain does not occur simultaneously on any two exposures.

On all fan systems there will be some loss of air in the distributing ducts. For this reason fans should be selected for about 5% to 10% greater capacity than the calculations actually show, depending, of course, upon the workmanship with which the duct systems are installed. For air conditioning work great care should be taken to insure tightness of the duct work, as excessive loss of refrigerated air is very expensive.

The field for air conditioning is still in a state of constant expansion, and much research work is in progress. World War II gave us many new applications for air conditioning, advanced engineering data and knowledge, and consequent improvements and refinements in methods of calculating air conditioning problems and in selecting equipment. New and interesting data can be found in current trade literature and technical journals. Such information is very helpful to the air conditioning engineer in keeping up to date on the changes and improvements in air conditioning apparatus, and on new and simplified methods for approaching the air conditioning problem.

Example II—Surface Cooling Calculations

Outside air quantity

$=1{,}170 \text{ people} \times 10 \text{ cfm} = 11{,}700 \text{ cfm}$

$$= \frac{11{,}700}{13.84} = 845 \text{ lb per min}$$

(Air quantity measured at inside conditions.)

Supply air quantity:

Internal sensible load

= total sensible − (fan motor and outside air heat gain)

$= 967{,}952 - (38{,}175 + 121{,}660)$

$= 967{,}952 - 159{,}835 = 808{,}117$ Btu per hr

Supply air 20 deg below room temperature

$= 80 \text{ F} - 20 = 60 \text{ F}$

Supply air quantity

$$= \frac{(808{,}117)(13.84)}{(0.24)(20)(60)} = 38{,}800 \text{ cfm}$$

$$= \frac{38{,}800}{13.84} = 2{,}800 \text{ lb per min}$$

Return air quantity

$= 38{,}800 - 11{,}700 = 27{,}100$ cfm

$$= \frac{27{,}100}{13.84} = 1{,}955 \text{ lb per min}$$

Entering air temp, db

$$= \frac{(1{,}955)(80) + (845)(90)}{2{,}800} = 83 \text{ F}$$

Entering air, enthalpy

$$= \frac{(1{,}955)(30.79) + (845)(34.04)}{2{,}800}$$

$= 31.8$ Btu per lb

∴ Entering air wb = 67.2F

Entering air rh = 44% (from Fig. 2)

Locate the above entering air conditions on the psychrometric chart, determining point A. This point should fall on the straight line between point B, the outside air condition of 90 db and 70 wb and the room condition of 80 db and 48% rh at C. Using the sensible-total heat ratio $h_s/h_t = 0.79$, construct line XY on the psychrometric chart from the reference point of 80 F db and 50% rh to 0.79 on the h_s/h_t scale. Draw another line ZA parallel to the first but going through the entering air condition of 83 db and 44% rh. The point where this second line intersects the saturation line at Z, determines the coil surface temperature (or the dew point if a washer is used), in this case 54 F.

$$\text{Leaving air db} = \text{Entering air db} - \frac{\text{Total sensible load in Btu per hr}}{(0.24)(\text{lb air per min})(60)}$$

$$= 83 \text{ F} - \frac{(967{,}952)}{(0.24)(2{,}800)(60)}$$

$$= 83 - 24 = 59 \text{ F}$$

The required leaving air condition will fall on the line ZA on the psychrometric chart. Therefore, refer to point D on Fig. 2 at 59 F leaving db temperature and find 56.5 F wb and 55.0 F dp as the air conditions leaving the surface cooling coils. One may now select direct expansion cooling coil from catalog.

Table 4 and equations 1, 2, and 3 are typical of direct expansion cooling coil rating data as published by various manufacturers.[12]

$$h_e - h_l = C_{vr}(h_e - h_r) - C_v(h_r - 6.89) \qquad (1)$$

$$L = 4.45v(h_e - h_l) \qquad (2)$$

Table 4. Table of Constants for Dry Expansion Coils

Air Velocity fpm	C_v	C_{vr}								
		Number of Rows								
		1	2	3	4	5	6	7	8	10
200	.540	.630	.800	.870	.920	.945	.965	.975	.983	.988
250	.435	.506	.660	.750	.830	.872	.906	.932	.953	.967
300	.361	.435	.587	.687	.770	.818	.861	.895	.925	.948
400	.271	.350	.500	.598	.683	.740	.790	.835	.874	.908
500	.216	.296	.436	.535	.617	.678	.733	.782	.825	.869
600	.182	.256	.387	.482	.563	.625	.680	.733	.778	.830
700	.155	.222	.344	.435	.513	.577	.632	.685	.733	.790
800	.135	.191	.302	.390	.470	.535	.585	.635	.687	.750
K_r		64	202	380	600	890	1300	1700	2175	3300
L_{min}		2740	5900	8150	10,000	11,370	12,620	13,760	14,700	15,700

$$t_a - t_r = \frac{L}{K_r + \frac{L}{20}} \qquad (3)$$

Where:

h_e = enthalpy of air entering coil, Btu per lb
h_l = enthalpy of air leaving coil, Btu per lb
h_r = enthalpy of air saturated at the temperature of the refrigerant, Btu per lb
6.89 = coil constant, Btu per lb
FA = face area of coil, sq ft
L = heat load on coil, Btu per sq ft FA hr
v = face velocity, fpm
K_r = coil constant, Btu per sq ft FA hr F
t_r = suction temperature of refrigerant, F
t_a = apparent surface temperature, F
4.45 = weight of dry air in a cubic foot of standard air times 60, lb

The selection of a coil from these data is a method of trial and error, as several selections might fit the requirements. The variables are refrigerant temperature, face velocity, and rows deep. In general, the refrigerant temperature should be selected from 5 deg to 15 deg below the coil surface temperature, and the face velocity between 450 and 550 fpm for coils operating dehumidifying. Air conditioning coils for comfort cooling are usually of staggered tube design, constructed of ½ in. OD or ⅝ in. OD tubing with 7 or 8 fins per inch.

The practical application of the typical manufacturers' data[12] to this example is as follows:

Cfm = 38,800
Entering air wb = 67.2 F Enthalpy = 31.8 = h_e
Leaving air wb = 56.5 F Enthalpy = 24.13 = h_l
required $h_e - h_l$ = 7.67 Btu per lb
Required coil surface temperature = 54 F = t_a
Total cooling load = 967,952 sensible
+254,980 latent
= 1,222,932 Btu per hr
= (L) (FA)
Try refrigerant temperature = 54 − 11
= 43 F = t_r
Enthalpy corresponding to t_r = 16.64 Btu per lb = h_r

Using equation 3

$$t_a - t_r = \frac{L}{K_r + \frac{L}{}}$$

$$54 - 43 = \frac{1{,}222{,}932}{\left[K_r + \frac{1{,}222{,}932}{(20)(\text{FA})}\right](\text{FA})} = 11$$

$$(\text{FA})\ (K_r) = \frac{1{,}222{,}932}{11} - \frac{1{,}222{,}932}{20}$$

$$= 111{,}169 - 61{,}147$$

$$(\text{FA})\ (K_r) = 50{,}022$$

Try 4 row deep coil: from Table 4, coil constant $K_r = 600$

$$\frac{50{,}022}{600} = \text{FA} = 83.4 \text{ sq ft}$$

$$v = \text{face velocity} = \frac{38{,}800 \text{ cfm}}{83.4} = 466 \text{ fpm}$$

$$\text{Required } L = \frac{1{,}222{,}932}{83.4}$$

$$= 14{,}700 \text{ Btu per sq ft FA hr}$$

The 4 row coil selected must now be checked by applying equations 1 and 2, and interpolating in table 4 for the constants C_{vr} and C_v, to see that the coil will remove the required amount of heat from the air.

$$(1)\quad h_e - h_l = C_{vr}(h_e - h_r) - C_v(h_r - 6.89)$$

$$= (0.648)(31.8 - 16.64)$$

$$- (0.235)(16.64 - 6.89)$$

$$= 9.8 - 2.3 = 7.5 \text{ Btu per lb}$$

$$(2)\quad L = 4.45v(h_e - h_l)$$

$$= (4.45)(466)(7.5)$$

$$= 15{,}600 \text{ Btu per sq ft FA hr}$$

This is about as close a selection as can be obtained by using standard cooling coils, and the total capacity at 43 F refrigerant temperature will be:

$$(L)\ (\text{FA}) = (15{,}600)(83.4)$$

$$= 1{,}300{,}000 \text{ Btu per hr} = 108 \text{ tons}$$

For such a large cooling load, six 18 ton coils would probably be selected. If each coil were made 30 in. high with tube length of 66 in., the total face area would be 82.5 sq ft which is close enough to the required face area. The actual face velocity would then be 38,800/82.5 = 470 fpm.

Example III—Dehumidifier Calculation

Outside air = 11,700 cfm = 845 lb per min
Supply air = 38,800 cfm = 2,800 lb per min
Return air = 27,100 cfm = 1,955 lb per min
Supply air dew point = 54F (See Example II)

It is impossible to obtain complete saturation in an air washer when dehumidifying. However, for practical purposes it is sufficiently accurate to assume 100% saturation for the dehumidifier calculations, in which case the dehumidifier leaving db = wb = dp = 54F.

Dehumidified air quantity

$$= \frac{\text{(Internal sensible + fan motor heat)}}{(60)(0.24)(80-54)}$$

$$= \frac{(808{,}117+38{,}175)}{(60)(0.24)(26)}$$

$$= \frac{846{,}292}{(60)(0.24)(26)} = 2{,}260 \text{ lb per min}$$

$$= (2{,}260)(13.84) = 31{,}300 \text{ cfm}$$

Bypass air = 38,800 − 31,300 = 7,500 cfm
All of the outside air should be dehumidified
Return air through dehumidifier = 2,260 − 845 = 1,415 lb per min
Entering air to dehumidifier:

$$\text{db} = \frac{(1{,}415)(80)+(845)(90)}{2{,}260} = 83.6\text{F}$$

$$\text{Enthalpy} = \frac{(1{,}415)(30.79)+(845)(34.04)}{2{,}260}$$

$$= 32.04 \text{ Btu per lb}$$

wb = 67.5F (from Table 1, Chap. 10)

dp = 59F = 74.54 gr per lb from chart of Chap. 10

	gr per lb
Moisture content at 59F entering dp	=74.54
Moisture content at 54F leaving dp	=62.01
Moisture removed in dehumidifier	=12.53

Total moisture removed

$$= \frac{(60)(2{,}260 \text{ lb})(12.53 \text{ gr})}{7{,}000 \text{ gr per lb}} = 243 \text{ lb per hr}$$

Latent heat equivalent
= (243 lb)(1,060) = 257,580 Btu per hr
Calculated latent load
(Example I) = 254,980 Btu per hr

	Btu per lb
Enthalpy at 70F outside wb	=34.04
Enthalpy at 54F dehumidifier wb	=22.59
Heat removed from outside air	=11.45
Enthalpy at 66F room wb	=30.79
Enthalpy at 54F dehumidifier wb	=22.59
Heat removed from return air	= 8.20

	Btu per min	tons
Total heat removed from outside air = (845)(11.45) =	9,675	
Total heat removed from return air = (1,415)(8.20) =	11,603	
Cooling load (sub-total)	21,278	= 106.4
Allow 10 hp for spray pump = 10 × 42.5 =	425	= 2.1
Total cooling load =	21,703	= 108.5

Select dehumidifier for 400 to 600 fpm face velocity

$$\frac{31{,}300 \text{ cfm dehumidified}}{600 \text{ fpm face velocity}} = 52.2 \text{ sq ft face area}$$

Allow 7 deg rise in spray water
Spray water required

$$= \frac{21{,}703 \text{ Btu per min}}{7 \times 8.33} = 372 \text{ gpm}$$

$$\frac{372 \text{ gpm}}{52.2 \text{ sq ft}} = 7.1 \text{ gpm per sq ft face area}$$

Using 2 spray banks

$$= \frac{7.1}{2} = 3.55 \text{ gpm per sq ft per spray bank}$$

54F dp − 2 deg difference = 52F leaving water
52F − 7 deg range = 45F entering water.

Difference between leaving air and leaving water depends upon dehumidifier type, and design values given are typical.

Thus one selects refrigerating unit and water cooler for 108.5 tons refrigeration when cooling 372 gpm spray water from 52F to 45F, using 2-bank air washer for 31,300 cfm.

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38. AIR SYSTEMS

1. The **performance requirements** of a distribution system are as follows:

a. To introduce to the conditioned space a quantity of air at a temperature and humidity which will maintain the desired indoor temperature and humidity.

b. To distribute the air so that:
   The temperature throughout the occupied zone is essentially uniform.

   The temperature difference between floor and breathing line is minimized.

   Motion of air will be adequate but not so great as to produce a sensation of draftiness.

c. To introduce the air as quietly as possible, so that noise from the air conditioning system will not be objectionable.

In addition the distribution system should be designed so as to blend with architectural treatment of conditioned space, be economical to install, and result in minimum owning and operating expense. **Fig. 1** shows typical situations.

2. Total air changes is defined as the number of complete changes per hour of the air in the room, giving the rate of circulation equal to the cfm discharged through the supply openings. This factor must not be confused with air changes equivalent to the rate of supply of outside fresh air. The total air changes of a given space

$$N = 60\,Q/V \qquad (1)$$

Where

N = Total air changes per hour

Q = cfm of air, referred to the specific volume at design conditions, equivalent to the total air discharged from the supply openings

V = Volume of conditioned space, cu ft

$$Q = NV/60 \qquad (2)$$

Uniformity of motion and temperature distribution are particularly dependent upon the relation of total air circulation to the volume of the conditioned space. With a small number of supply openings and a low total of air changes, there may be a tendency toward dead air pockets. On the other hand a high number of changes may mean drafts.

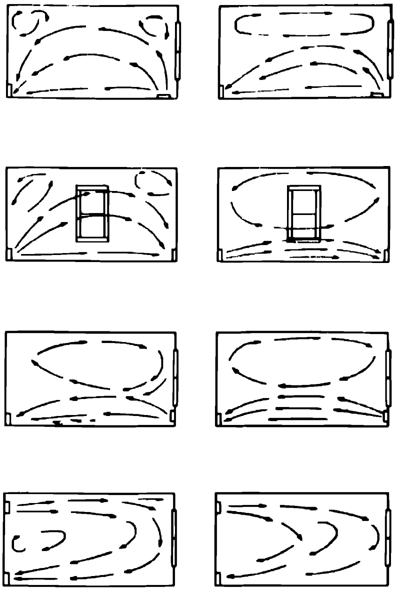


Fig. 1. Vertical Sections of Small Rooms Showing Types of Circulations

Table 1 may be used in estimating the **quantity** of air supply, or in determining whether a given rate of supply will give satisfactory air distribution. Judgment must be exercised in using this table, as it sets up wide limits.

If the number of air changes is approximately halfway between the upper and lower limits given in Table 1, no particular difficulties should be encountered. Rates of air changes within the lower portions of the ranges given are more suitable if the heat loss or heat gain per unit volume is low than if the heat loss or gain per unit volume is high.

3. Air changes in excess of 12 per hour generally are suitable only when the heat gain per unit volume is very high. In such cases, every possible effort should be made to reduce the **internal sensible heat gain** by insulation, awnings, use of mechanical ventilation, Venetian blinds, etc. When it is not possible to do this, there should be a compromise between the number of air changes and diffusion temperature difference. The **temperature difference** is the temperature difference between the air temperature at supply opening and design indoor temperature. This diffusion temperature difference should be increased if possible. Relocating or increasing the number or decreasing the size of outlets may permit this.

When the number of air changes is very high, the air should be directed toward unoccupied spaces, and induced secondary circulation should be avoided. With 12 or more changes per hour the air in the room is pretty certain to be well stirred up, and more effort should be directed toward avoiding drafts than toward supplying the conditioned air to the exact sources of heat gain. In all types of applications with very **high ceilings** the number of air changes should be based on the volume from the **floor level to approximately six feet above the level of the supply openings, neglecting** the remaining volume. Unless the number of air changes is very high this upper portion of the space will remain relatively undisturbed.

4. The **temperature** and **quantity** of air introduced to the conditioned space are dependent on each other, with heat loss or heat gain as the relating factor. The formula for sensible heat is

$$h_s = \frac{60Q}{v} \times 0.24 \times \text{dtd} \qquad (3)$$

From this we get the following formulae for air quantity required to accomplish a given heating or cooling load.

$$Q = \frac{h_s}{1.06 \text{ dtd}} \text{ for cooling} \qquad (4)$$

$$Q = \frac{h_s}{1.08 \text{ dtd}} \text{ for heating} \qquad (5)$$

where

Q = Total air supply, cfm referred to design indoor conditions

dtd = Diffusion temperature difference, deg F

h_s = Internal sensible heat gain for cooling, and internal sensible heat loss for heating, Btu per hr

v = specific volume of air at design outdoor conditions, cu ft per lb

Internal heat loss or heat gain refers to

Table 1. Approximate Rates of Air Supply for Heating and Cooling

Type of application	Location of supply openings	Approximate equivalent air changes per hour	
		Heating	Cooling
Residences	In baseboard near floor, discharging horizontally	3–6	—
	In floor, discharging vertically upward	3–6	6–9
	High in wall, discharging horizontally	5–9	6–9
Offices, stores, and restaurants	Generally high in wall, discharging horizontally	5–8	6–12
Theaters, assembly halls, dance halls, etc.	Generally high in wall, discharging horizontally	5–10	6–12

the heat loss or heat gain exclusive of that of ventilation air which is mixed with recirculated air before entering the air conditioner. In the above formulae the constants 1.06 and 1.08 are determined so that Q is the cfm based on the specific volume of air at design indoor conditions. Values corresponding to 80 F, 50% relative humidity for summer and 70 F, 35% relative humidity for winter, are used.

5. In order to obtain the maximum diffusion temperature difference in the above formulae for cooling, refer to **Table 2.** It is evident that greater diffusion temperature differences are allowable when many discharges are used than with few. Again, as the height of the discharge increases, the allowable difference increases.

Table 2. Approximate Maximum Diffusion Temperature Differences Permissible for Various Heights of Supply from Floor (for Cooling)

Height of supply above floor, ft	Maximum diffusion temperature difference, deg F	
	Relatively few large volume discharges	Relatively many small volume discharges
10	15	20
12	17	22
14	19	24
16	21	26
18	23	28
20	25	30

If the temperature difference between room air and supply air is increased, the air stream bends down more steeply and tends to enter the occupied zone a shorter distance from the point of supply. Hence, for any given height of grille there is a **minimum height of supply** above the floor for a given discharge velocity and type of grille. If the grille is located at lower level, air will enter the occupied zone at an undesirably high velocity and possibly at too low a temperature.

The diffusion temperature difference, on heating, should be kept as low as possible. However, if the discharge temperature is less than about 100 F, great care should be taken to prevent this air from blowing directly over an occupant. Usually 100 F is as low as warm air should be discharged into a room. Air at lower temperatures seems cold to the average person if the velocity is greater than 40 to 50 fpm.

To summarize, in designing air distribution systems it is necessary to consider both diffusion temperature difference and total air changes in relation to the type and location of supply openings. The total air changes must be adequate to give good distribution and at the same time the temperature difference must be low enough so the air stream will not enter the occupied zone before its temperature has been raised essentially to that in the room or before its velocity has been reduced to approximately 50 fpm.

Gravity Flow of Air

6. When air is heated it tends to rise and when cooled it tends to fall. Supply and return **openings** should be located with due consideration for the natural gravity flow characteristics of warm and cool air. For **heating,** this would indicate that warm air supply openings should be located near the floor blowing horizontally, or near the ceiling blowing slightly downward. Similarly for **cooling,** the air should be introduced horizontally through openings well above the floor level, or vertically upward through openings near or in the floor.

If a given cooling load is to be handled with a relatively small quantity of air, the diffusion temperature difference will be large, and care must be exercised to prevent cold air from falling too quickly to a level where it can produce a sensation of draftiness. The cold air must be warmed by mixing action, which may be promoted by:

a. Locating the supply openings so that the supply air is in contact with room air for a longer time before reaching the occupied zone. This may be accomplished by high outlets, high discharge velocities, or directing the air slightly upward.

b. Breaking up the supply air into a number of relatively small jets so that mixing takes place rapidly. For example, numerous small-volume, well-distributed outlets or long, narrow, slot-type outlets with directional flow.

c. Use of special grilles such as aspirating, directional flow, etc.

7. Air may be **projected** to a desired point by giving it adequate velocity and proper direction at point of supply, but this method of control of air motion is not applicable in the case of **return openings.** Tests have shown that even with a face velocity of 1,000 fpm the air velocity is not more than 50 at distances more than 3 ft from a return grille. Thus returns must be located so that air will flow to them freely by gravity.

Returns for heating should be located so that air does not travel a great distance nor encounter a number of bends. Cold air comes from the point where heat loss occurs, such as windows and outside walls. In cooling the location of returns is less vital. In theory they should be located well above the floor but in practice gradients in cooling are not so great as in heating.

The **number of returns** is variable. Relatively few are sometimes adequate in both heating and cooling systems, but in cases of highly concentrated loads or bad exposure, many well-distributed returns are often required.

8. As a jet of air issues from an opening it encounters **resistance** to flow, partially by friction between the air in the sides of the jet and the air in the room. The amount of room air thus **entrained** increases with the perimeter and velocity of the jet. At relatively high initial velocities (800 fpm or more) the amount of entrained room air may be great enough to produce a very appreciable secondary or induced circulation. This entrainment of room air may be increased, at a given velocity, by using grilles specially designed to increase turbulence in the air jet (**aspirating grilles**). On the other hand, the ratio of perimeter to area may be kept small, to reduce the amount of entrained room air and thus to maintain the jet velocity.

The secondary circulation induced by the entrainment of room air is often a great aid in maintaining uniform air motion and temperature distribution, with a minimum number of supply openings and with a minimum quantity of air. The entrained room air also tempers the supply air, reducing the tendency for cool air to fall to the floor and room air to rise to ceiling.

Grille Characteristics

9. Air flow characteristics of different types and arrangements of supply openings and grilles must be considered in conjunction with the principles discussed above. The important characteristics to be considered may be enumerated as follows:

a. **Throw** vs. proportions and arrangement of the grille, and discharge velocity. This determines the distance conditioned air may be projected from the point of supply.

b. Horizontal **spread** of the air vs. proportions of the grille. This affects the number of openings required to cover a given area.

c. **Air velocity** at the lower edge of the air stream vs. proportions and arrangements of the grille, diffusion temperature difference, discharge velocity, horizontal distance from grille and vertical distance from grille level. Cool air discharged from a grille drops, causing the air stream issuing from a supply opening to describe a path which bends downward. Furthermore, the velocity of the air at the boundaries of the air stream is lower than the average velocity. It is important that the air velocity should not be in excess of 40 to 50 fpm where the air stream enters the occupied zone.

d. Vertical **fall** or rise of air from the supply opening to where the jet temperature is essentially equal to that in the room vs. proportions and arrangement of the grille, diffusion temperature difference and discharge velocity. This is useful in determining whether a given arrangement will spill cold air into the occupied zone in summer, or cause stratification of warm air at the ceiling in heating; hence it is practicable in designing a system to avoid these two undesirable effects.

e. **Quantity of entrained room** air as a percentage of the initial jet volume vs. proportions and arrangement of the grille, and discharging velocity.

Data concerning the above characteristics are available in varying degrees of completeness from grille manufacturers.

10. The following conclusions on **air distribution** are based upon experimental work conducted by one well known grille manufacturer:

a. The throw from a straight flow grille varies with the square root of the daylight area of the grille and with the face velocity within ordinary grille proportions.

b. The aspect ratio of a grille has no appreciable effect on the distance of air throw.

c. Fanning out the air stream shortens the throw, the amount depending on the degree of deflation.

d. The drop, for a given throw, of an air stream below room temperature, varies about inversely as the face velocity and directly as the temperature differential.

11. Tables 3 and 4 may be used as guides in estimating **discharge velocities** for bar-type or lattice-type grille without deflectors, and for plain nozzles. These tables are approximate, and intended only for use in the general preliminary layout. The performance of numerous special types of grilles may differ appreciably from these tables.

When cooling with floor openings discharging vertically, the velocity depends upon the ceiling height. A velocity of at least 500 fpm should be used for average ceiling heights of 7 to 8 ft. For higher ceilings the velocity should be increased somewhat. When heating with floor openings discharging vertically, the velocity should be kept low, not over 400 fpm if possible. Velocities of approximately 500 fpm should be used only when the system is to be used for both heating and cooling, with the same air quantity the year 'round.

Table 3. Approximate Velocities for Various Diffusion Temperature Differences and Lengths of Throw

Throw from point of supply, ft	Approx velocity over face area, fpm			
	Diffusion temperature difference, F			
	15	20	25	30
15	400	525	650	775
25	500	625	750	875
35	600	725	850	975
50	750	875	1000	1125
75	1000	1125	1250	1375
100	1250	1375	1500	1625

Table 4. Approximate Supply and Return Air Velocities

Type of opening	Average velocity through free area of grille, fpm	
	Heating	Cooling
Supply—Low in wall, discharging horizontally	300-400	—
Supply—In floor, discharging horizontally	300-400	500-800
Supply—High in wall, discharging horizontally	600-1,500*	300-1500*
Return	300-500	300-500

* Note: Higher velocities are sometimes permissible with some types of supply grilles, and when a long throw of the air is required.

Return velocities given in Table 4 represent average practice for residential applications, where it is usually necessary to do everything possible to keep the resistance of the duct system low. However, in commercial applications higher velocities (up to 1,000 fpm) may be used.

Certain precautions must be observed in installing high-velocity supply systems. There should be no obstructions such as columns or chandeliers in the direct path of the air jet. If the ceiling has cross beams perpendicular to the direction of the air jet, the supply openings should be located at least 2 ft below the lowest point of the ceiling.

Location of Openings in Relation to Exposure

12. A considerable percentage of the conditioned air should be supplied to that portion of the space where heat gain or heat loss occurs. For heating the best way to accomplish this is to discharge the warm air across the exposed walls. In cooling the problem is more one of a fairly uniform distribution of supply air throughout the conditioned space. The following summary, coupled with experience and judgment, may be used as a guide in locating supply and return openings. In general, supply openings are to be located to minimize the possibility of blowing air directly on occupants; return openings near unusual sources of odor concentration should be avoided; exhaust vents to outdoors are advisable where there are sources of exceptional odor concentrations; supply risers should be placed in inside partition walls where possible to avoid excessive heat loss in duct and low discharge temperature.

With respect to supply openings in cooling systems, the following requirements apply:

a. **High in wall blowing horizontally:** This arrangement has proved most satisfactory where air is discharged at velocities of 500 fpm or more. Care should be exercised in using outlets near ceiling to prevent streaking. Supply outlets should be about 7 ft from floor and have air directed horizontally.
b. **Ceiling outlets** with diffusers or a deflecting pan to direct the air horizontally are very satisfactory. In places where ceilings are very low, ceiling outlets can be used to advantage to avoid drafts.
c. **Floor openings along outside wall blowing upward** give reasonably satisfactory results, with discharge velocity of 500 fpm or more, especially in residences, as a compromise arrangement for summer and winter operation. The air should be deflected at a slight angle from the vertical, to prevent the air from mushrooming, i.e., falling directly down on itself.
d. **Baseboard supply** is unsatisfactory for cooling, if air is directed horizontally. There are baseboard registers now made that are adjustable, such that for cooling operation the face damper is closed and the top of the grille pulled out so that the air is discharged vertically.

13. In large commercial jobs it is almost always necessary to have some form of **zone control** in order to maintain even temperatures in the different conditioned spaces, because of different and changing load conditions for both heating and cooling. Some of the most common methods of zoning are as follows:

a. Separate conditioner, fan, and duct system for each zone.
b. Central conditioner and fan with separate branch duct systems for each zone, controlled by automatic dampers in each zone.
c. Central conditioner with blow-through fan, where zones are divided at the coils in the conditioner with separate duct systems in each zone. This type of system usually has automatic control of the individual coils in each zone (see p. 648).
d. Central conditioner and fan for entire system, with booster coils in each zone duct system automatically controlled from each zone.
e. Central conditioner and fan for entire system with booster fans for each zone automatically controlling the air flow to each zone. Sometimes this type of zone control uses booster coils in each zone.

Air Filters

14. Air cleaning is the removal of dust and dirt from the air, which may be accomplished in several ways:

a. By passing the air over a surface coated with an adhesive material which literally "catches" dust particles carried by the air.
b. By passing the air through a finely woven fabric, which "screens" dust particles from the air.
c. By changing the direction of the air abruptly, thus "throwing" the dust particles out of the air stream.
d. By passing the air through a water spray. The water droplets collect on dust particles and cause them to drop down out of the air stream.
e. By passing the air between charged electrodes, which induce electrical charges on the dust particles and cause them to be precipitated from the air stream.

Each one of the above methods may be

used singly, or in combination, in various applications.

Viscous air filters employ the first method listed above. In the unit the filtering material is arranged in units of such size as to be convenient for installing and servicing. Each unit is a pad of filter material (such as spunglass wool, steel wool or animal hair), and the necessary supporting frame, arranged so that it can be removed easily for cleaning or replacing. Unit filters are made of hemp or paper fiber to be thrown away after using.

The **automatic type** of viscous filter uses the same principle of filtering as the unit type, but the filter is kept clean automatically. One method of accomplishing this is to support the filter material on an endless belt which is moved slowly, or intermittently, through a bath of the fluid.

Dry air filters make use of the second method of air cleaning listed above. Such filters are always of the unit type, and may be either cleanable or throw-away, more frequently the latter.

Scrubbers are more effective as filters than air washers. They make use of the third method of air cleaning listed above. The zig-zag path of flow throws the air against the baffle plates. Some of the dust particles are thus collected on the wetted surfaces and flushed to the collecting basin. The scrubber has the further advantage over the air washer in that the temperature and humidity of the air are not greatly altered.

15. **Air washers and humidifiers** may be classified in two distinct types; (a) the spray type and (b) the surface type. Dehumidifiers may be classified in three types; (1) spray, (2) surface, (3) chemical dehydration types.

Air washers are usually of the spray type, and if properly designed and adapted may serve as a humidifier, dehumidifier, or both. The spray-type washers commonly used for air conditioning and ventilation are made in standard designs and sizes to meet the requirements of a given duty. The classification of design is usually expressed in the number of banks of spray nozzles, of which there are never more than two in a standard unit.

The two standard designs are **one-bank** and **two-bank** washers. The banks of nozzles referred to are spray nozzles, which spray the water in fine particles and distribute the spray over the entire cross section of the washer chamber. All washers are usually equipped with one extra row of flooding nozzles which spray a sheet of water down the entering side of the leaving eliminator plates to wash down any dirt and dust which may have accumulated and to help remove some of the entrained moisture being carried through the washer by the air.

Tables 5 and 6 give typical specifications for one-bank and two-bank washers, devices arranged with flooding nozzles to wash the eliminator plates. A one-bank washer is satisfactory for humidifying and washing the air, but is not adequate for cooling or dehumidifying.

The quantity of water sprayed in a standard one-bank washer varies from 4 to 5 gal per 1,000 cu ft of air based on an air velocity of 500 fpm through the washer chamber. The flooding nozzles spray 3 to 4 gal of water per min per ft of washer width.

The **saturation efficiency,** or the reduction of the entering wet-bulb depression, of a one-bank washer ranges from 60 to 70%, depending upon the fineness of the spray, distance between the spray and eliminators, and velocity of air through the washer chamber.

The spray nozzles in a two-bank washer spray the same quantity of water as those in a one-bank washer, but as there are twice as many, the quantity of water sprayed would range from 8 to 10 gal per 1,000 cu ft of air based on an air velocity of 500 fpm through the washer chamber. The saturation efficiency of a two-bank washer ranges from 85 to 95%.

The **resistance** to air flow of either the one-bank or two-bank washer ranges from 0.2 to 0.4 in. of water, depending upon the spacing and construction of the eliminators, type of entering baffles, direction of sprays (either with or against the direction of air flow), and the connections to and from the washer which may vary the air velocity across the face of the washer and thus cause eddy currents in the spray chamber.

Details of **construction** of air washers are practically the same for all makes of stand-

Table 5. Typical Specifications of One-Bank Washers

Number	Air capacity, cfm	Chamber area, sq ft	Dimensions, in. W	H	L	Washing surface, sq ft	Number of nozzles Spray	Flooding	Water flow, gpm Spray	Flooding	Pump size, in.	Motor hp
1	4,600	9.2	33	61$\frac{1}{2}$	86$\frac{1}{16}$	84.5	12	9	22	9	1$\frac{1}{2}$	1$\frac{1}{2}$
2	9,400	18.8	64$\frac{1}{2}$	61$\frac{1}{2}$	86$\frac{1}{16}$	174	24	20	43	20	1$\frac{1}{2}$	2
3	14,200	28.3	95$\frac{3}{4}$	61$\frac{1}{2}$	86$\frac{1}{16}$	263	36	30	65	30	1$\frac{1}{2}$	3
4	21,900	43.7	95$\frac{3}{4}$	61$\frac{1}{2}$	86$\frac{1}{16}$	413	60	30	108	30	1$\frac{1}{2}$	3
5	29,300	58.6	127$\frac{1}{4}$	61$\frac{1}{2}$	86$\frac{1}{16}$	553	80	40	144	40	3	5
6	36,700	73.4	158$\frac{1}{2}$	61$\frac{1}{2}$	86$\frac{1}{16}$	693	100	51	180	51	1$\frac{1}{2}$	5
7	44,100	88.2	190	61$\frac{1}{2}$	86$\frac{1}{16}$	852	120	61	216	61	1$\frac{1}{2}$	5

Table 6. Typical Specifications of Two-Bank Washers

Nominal size, ft	Air capacity at 500 fpm, cfm	Area of spray chamber, sq ft	Dimensions, ft-in. W	H	Elimination surface, sq ft	Spray nozzles, no.	gpm	Flooding nozzles, no.	gpm
5 ×4	10,000	20	5-3	5-1	244	80	9	18	98
5 ×6	15,000	30	5-3	7-1	366	120	9	18	138
5 ×8	20,000	40	5-3	9-1	488	160	9	18	178
5 ×9	22,500	45	5-3	10-1	549	180	9	18	198
6 ×5	15,000	30	6-3	6-1	366	120	11	22	142
6 ×7	21,000	42	6-3	8-1	512	168	11	22	190
6 ×9	27,000	54	6-3	10-1	658	216	11	22	238

ard air washers, the only difference being in the type and capacity of nozzles, entering baffles, and detail of design of the leaving eliminators.

The **casing** is constructed of either 18-gage galvanized steel or 30-oz copper, rigidly braced with galvanized iron or brass angles to prevent sagging and "breathing" of the casing. All angle bracing and door frames are riveted with rivets soldered. All seams are soldered, and riveted with rivets soldered. The **tank** is constructed of 14- to 16-gage galvanized steel or 36- to 40-oz copper. The tank is also rigidly braced with iron or brass angles for stiffening. All angles and seams are riveted and soldered except in larger sizes where tanks are welded.

Air washer **nozzles** differ in design and performance with each manufacturer. Capacities of the nozzles range from .9 to 2.0 gpm, each with pressure ranging from 17 to 30 psi. They are uniformly spaced both vertically and horizontally over the cross section of the washer to give a uniform distribution and density of spray.

39. SMALL COMFORT COOLING SYSTEMS AND UNITS

THERE are four different methods of comfort cooling: (1) Mechanical refrigeration; (2) cold water; (3) air circulation and ventilation; and (4) evaporative cooling. Each method will be discussed from the viewpoint of its advantages and limitations.

Mechanical Refrigeration

1. The most widely used of the four methods is mechanical refrigeration. It is the only one which provides positive means of control to obtain comfort cooling. There are a number of variations in the construction and installation of this type of equipment, which may be classified as follows:

a. A self-contained unit within a room
b. A self-contained unit within a room, using ducts and grilles for better air distribution
c. Cooling section within a room (with or without ducts), connected to a remote compressor
d. A unit installed outside of the conditioned area, using ducts and grilles.

2. At the present time there is a growing preference for **self-contained units** covering small comfort cooling systems. This type of unit has a definite place in small stores, restaurants, homes, apartments, single rooms, etc. The reasons for its increasing popularity can be listed as follows:

a. Made on production basis; low initial cost
b. Low installation cost; completely assembled at the factory, thus reducing installation to a minimum
c. Attractive finish permits mounting within the conditioned area
d. Complete range of sizes available
e. Isolated rooms may be treated
f. Units are virtually portable.

A self-contained unit consists of the following parts contained in a single cabinet: compressor, condenser, receiver, direct expansion coil, fan, filter and controls. The unit can cool, dehumidify, filter and circulate air. In many cases ventilating, heating and humidifying are also available.

Self-contained units are constructed with both air and water-cooled **condensers.** The air-cooled condenser requires a large quantity of air that must be picked up out-

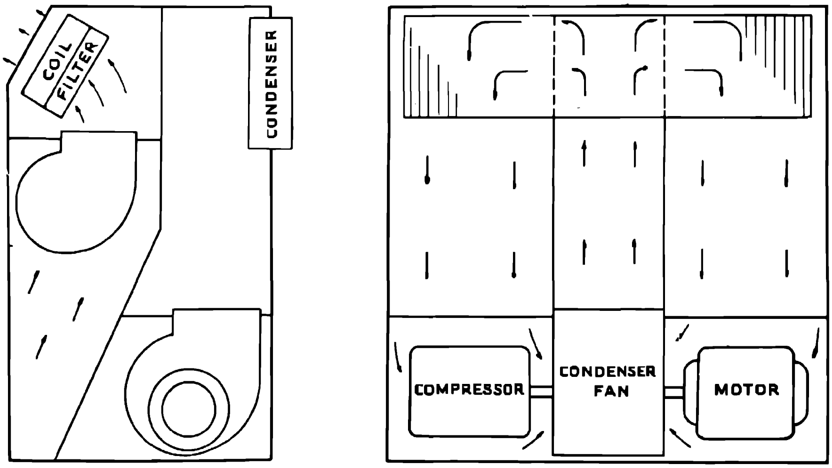


Fig. 1. Floor Type of Self-Contained Unit (Air-Cooled Condenser)

side of the conditioned area, circulated through the high side, and discharged out of the area. It is necessary that this type of unit be installed at an outlet such as a window. Due to this limitation, the air-cooled, self-contained unit has been limited to the ⅓, ½ and ¾-hp sizes. These units are generally referred to as room coolers.

3. There are two general types of **room coolers**, floor and window models. In the floor model (**Fig. 1**) all the mechanism is inside the room and the condenser air circulates through a short duct at the top rear of the unit. This type of unit will take some floor space but requires approximately 6 in. of window space for condenser air. The window model is a complete unit placed in a window (**Fig. 2**). The low side (evaporating coil, circulating fan and filter) is usually on the inside of the room, while the high side (compressor, condenser and condenser fan) usually extends outside. Many designs have attempted to minimize the height of the window type, since a high unit cuts out a large portion of the window, reducing its lighting effect.

Since the room cooler is designed for free air delivery, ducts should not be used. Dampers are usually provided for fresh air intake or exhaust discharge. The condensate from the cooling coil is disposed of by evaporation, using the condenser air; this eliminates the need of a drain. Upon installation a unit of this type requires only an electrical outlet. The normal control is by a manual switch with no provisions for thermostatic control with the unit. Most units are wired to operate the ventilating fan with or without a compressor.

Room coolers can be designed to incorporate **heating** by using the **reverse cycle** principle. Such a unit can provide heat for a single room in the spring or fall when the main heating plant is shut down. Since the refrigeration system is fundamentally a heat pump, removing the heat from the inside and dissipating it to the outside, by reversing the cycle (that is, making the cooling coil the condenser and the con-

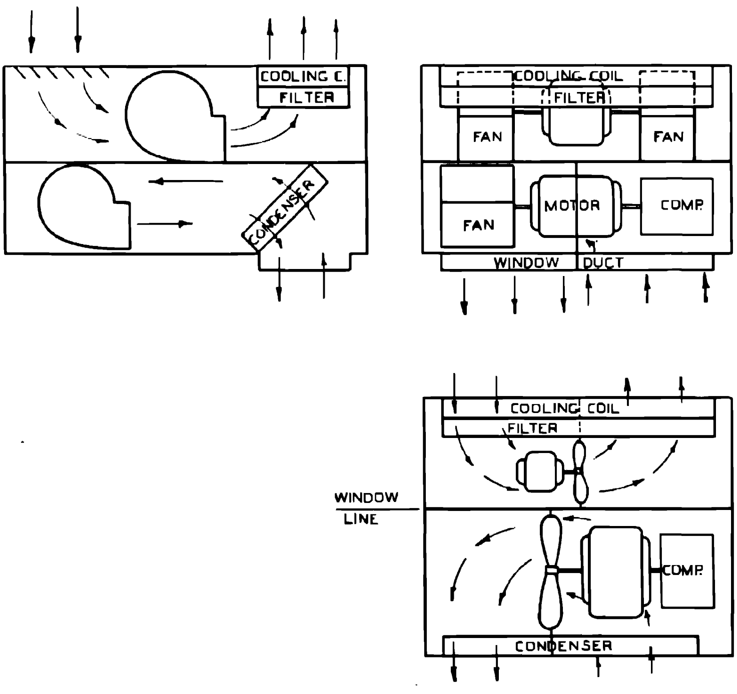


Fig. 2. Self-Contained Units

denser doing the work of the evaporator) heat can be pumped from the outside air into the room. The heat thus dissipated would be two to four times the electrical energy input. The reverse cycle is limited in usefulness to an outside temperature of approximately 45 F.

4. Self-contained units using water-cooled condensers are usually constructed in sizes of 1 hp or larger, and are commonly referred to as **store coolers.** These units come in a range of sizes as follows: 1, 2, 3, 5, 7½, 10, and 15-hp. The most common are the 3 and 5-hp. It is difficult to distribute satisfactorily large volumes of conditioned air from one outlet; thus store coolers larger than 5 hp are seldom used without ductwork.

The larger self-contained unit (**Fig. 3**) is a complete air conditioning unit designed for installation within a conditioned area. It provides summer air conditioning, that is, cooling, dehumidifying, filtering, ventilation and air circulation. Winter air conditioning, heating and humidifying, can be provided if desired.

The complete high and low side are contained within the store cooler. All electrical work is completed and ready for operation. The only connections required are electrical and water service and sewage. The air intake and discharge grilles are arranged so that they can be mounted on the front or back of the unit as desired for each individual job. The discharge grille is provided with adjustable louvers.

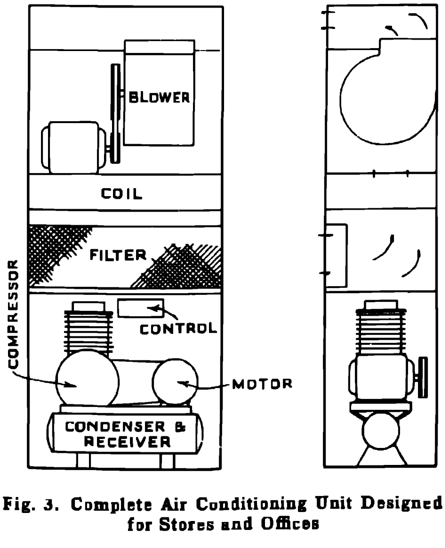


Fig. 3. Complete Air Conditioning Unit Designed for Stores and Offices

When **ducts** are used the increased static pressure can be compensated by increasing the fan speed by means of an adjustable drive pulley on the fan motor.

A **control** switch is provided with "off," "ventilating" and "cooling" positions. When the switch is on the "ventilating" position, the fans provide circulation of air but the compressor is not in operation. At the "cooling" position, the fan and compressor are in operation. The unit may be provided with a thermostat in the air stream and an adjustable control knob mounted on the outside of the unit. If a remote-type thermostat is required, it can be incorporated.

Means of removing motor heat, and the heat of compression, must be provided. This is usually accomplished by passing the condenser water through a small finned cooling coil placed within the compressor compartment. The water passes through the cooling coil before it enters the condenser. Since the larger self-contained units are usually placed outside the conditioned area, the enclosure around the condensing unit is screened or perforated to allow circulation of air.

A Standard (ASRE) Code for testing self-contained units (page 678) has been adopted. Most manufacturers have rated their units under the code conditions. For very rough estimating, the value of one ton per hp is sometimes used, but for exacting comparative values reference should be made to the manufacturer's code ratings.

5. In a **semi-remote installation** the low side is placed within the conditioned area, while the compressor is placed in a remote position. The range of sizes in this type of equipment is usually 1 to 10 tons. It is necessary to run refrigeration leads and a condensate drain from the unit. The suction line and drain must be insulated to prevent dripping on a humid day.

This setup has an advantage over the self-contained unit; since the compressor is remote, the noise level will be lower. These units are usually of the ceiling type which requires less valuable space than the floor unit. An evaporative condenser can be used if desired. A semi-remote unit is seldom used with ductwork, but if a pressure-type fan is used, ductwork can be applied.

6. In a remote installation (Fig. 4) both the high and low side are located outside the conditioned area, with ducts and grilles distributing and circulating the conditioned air. The compressor may be mounted in any convenient place, and an evaporative condenser may be used. This type of equipment usually ranges in capacity from 3 to 25 tons, as indicated in **Table 1**. Units having a capacity up to 50 tons can be obtained. The self-contained unit discussed previously may be used for remote installations.

Although remote installations may prove more expensive, they have definite advantages. Since all the air conditioning equipment is outside the conditioned area, the noise level is at a minimum. Better air distribution can be obtained with ducts and grilles and the ducts can be blended into an architectural design.

It is difficult to determine the exact dividing line between the use of self-contained units and remote installations. The following factors will influence the final decision: Noise, cooling load, size and shape of the floor area, ventilation requirements, decoration problems, initial equipment cost, installation and operation costs.

Cold Water Units

7. The use of cold water for a small air conditioning installation is not common, since it requires a large quantity of water. The water temperature must be below 60 F to be effective. To find the quantity of water required, the following formula can be used: gpm × temperature rise of water × 500 = Btu per hr.

As an example, let us assume a load of 5 tons (60,000 Btu per hr) with 50 F water and a 6 deg rise in water temperature. The gpm required will be 60,000/(6 × 500) = 20. The capacity of a cooling coil depends upon the coil temperature. Therefore the size of the coil for a given gpm and capacity will depend upon the amount and temperature of the water.

The **cooling unit** using cold water as a cooling means is similar to the low side of a mechanical refrigeration system. It consists of the following **elements**: Fan and motor, filter and water coil, encased within a metal shell. Three water connections are required—water feed, water return, and condensate drain. In units above 15 tons in capacity, a water spray system is sometimes used, but for smaller systems a water coil is used. The unit is made up as a ceiling or floor type, the ceiling unit being more popular. It may be mounted within the conditioned area or in a remote position.

8. In all air conditioning jobs **fresh air** is desired, and in most instances it is a necessity. The amount varies with the type and size of installation. For values, see page 618 covering load calculations. The

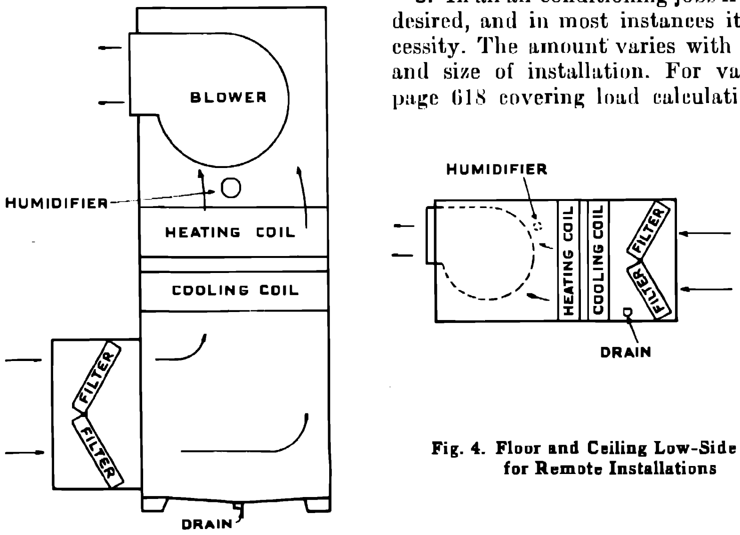


Fig. 4. Floor and Ceiling Low-Side Units for Remote Installations

Table 1. Typical Ratings of Low-Side Units

(Refrigerant at 40 F, entering air at 80 F db and 67 F wb)

Unit number	Air volume, cfm	Coil face area, sq ft	Number of stacks	Capacity, Btu/hr	S.H.R.[1]	Leaving temp, F	
						db	wb
1	1,200	3	2	27,200	.68	66	60.0
			3	37,300	.66	61	57.1
			4	46,000	.65	57	54.5
			5	53,000	.64	54	52.3
			6	59,000	.63	52	50.4
2	2,000	5	2	45,300	.68	66	60.0
			3	62,400	.66	61	57.1
			4	76,600	.65	57	54.5
			5	88,500	.64	54	52.3
			6	98.300	.63	52	50.4
3	3,000	7½	2	68,000	.68	66	60.0
			3	93,400	.66	61	57.1
			4	115,000	.65	57	54.5
			5	132,500	.64	54	52.3
			6	147,500	.63	52	50.4
4	4,000	10	2	90,500	.68	66	60.0
			3	124,300	.66	61	57.1
			4	153,000	.65	57	54.5
			5	177,000	.64	54	52.3
			6	196,500	.63	52	50.4
5	6,000	15	2	136,000	.68	66	60.0
			3	187,000	.66	61	57.1
			4	230,000	.65	57	54.5
			5	265,000	.64	54	52.3
			6	295,000	.63	52	50.4

[1] S.H.R. = Sensible heat ratio = $\frac{\text{Sensible heat}}{\text{Total heat}}$

inlet must be so located as to insure the delivery of fresh, clean air, and should be away from the ground to minimize dirt or dust. This intake must be protected by louvers and a wire bird screen and joined to the unit by means of a canvas connection.

9. When ducts are used, care must be taken to keep the **noise level** below an objectionable point. The movement of air through a duct produces some noise, which varies as the velocity of the air. If the sound caused by the air in the duct is below the sound level of the conditioned area, no air noises will be heard. Where there is already some noise, as in restaurants and busy shops, the duct velocity should be designed for 1,000 to 1,100 fpm. Where the noise must be at a minimum, however, the maximum allowable is 650 to 750 fpm. If additional noise reduction is desired, the ducts may be lined with acoustical material.

If the duct is connected directly to the unit, making a metal-to-metal contact, any **vibrations** set up within the unit will be transmitted to the duct and be objectionable at a distant point. To prevent such transmission of vibration, a canvas collar should be connected close to the unit, with one or two inches of slack. The canvas should not be painted. Complete data covering ducts and their application can be found in Chap. 15.

10. In order to permit easy selection by untrained sales personnel, many manufacturers have developed forms for rapid **calculations of heat loads.** These methods, based on average conditions, will in general give satisfactory results, but if there are

any unusual conditions the results are misleading. For accurate load calculations see Chap. 37.

An **undersize** unit should not be used with the idea of getting a little cooling effect. The result may be less comfortable than without a unit with the windows open to the breeze. It is not practicable to cool an adjacent room through an open door.

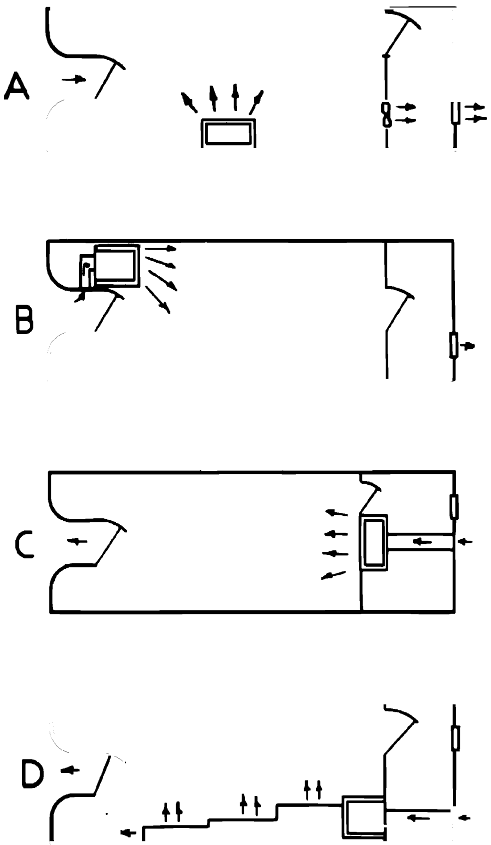


Fig. 5. Typical Air Introduction Methods

Fig. 5 illustrates a number of methods of cooling a given area. Method A has no ducts, fresh air being obtained by means of an exhaust fan. A self-contained or a semi-remote unit may be used. Method B is the same, except that there is a positive means of securing fresh air.

Method C is a remote installation requiring a minimum of ducts; it uses a single outlet and a single return. A self-contained unit or a built-up unit consisting of a separate low and high sides can likewise be used. Method D is similar except that ducts and grilles are used, resulting in even air distribution, with the minimum of draft. If the size of the room warrants it, a return air duct can be used.

Air Circulation and Ventilation

11. The **attic exhaust fan (Fig. 7)** has proved popular as an inexpensive means of obtaining cooling effect in a home. While it cannot be called a substitute for complete air conditioning, its principle is sound.

The maximum temperature during the day usually occurs between noon and 4 p.m. During this time a considerable amount of heat is stored up in the construction material of the building. If the air in the attic is not circulated, temperatures as high as 130 F are common. After sunset the outdoor temperature begins to drop, with the lowest temperature occurging between midnight and 6 a.m. The difference between the maximum and minimum temperatures during the day varies from 15 to 20 deg F. The heat that is stored in the building materials is slowly dissipated during the night.

The attic fan should be started after sundown to draw cool air from the outside and circulate it through the house. The indoor temperature is lowered quickly and the movement of air also has a cooling effect on human bodies within.

There can be no set method of adjusting and controlling the air flow as each house differs in construction. It is advisable in those rooms to be cooled to open the windows most distant from the fan. It may be necessary to close or partly close other windows.

Care must be exercised in selecting the proper fan size. To obtain best results a fan should be selected to give one air change of the entire area in two to four minutes. This quantity of air will be sufficient to cause a gentle breeze through the area.

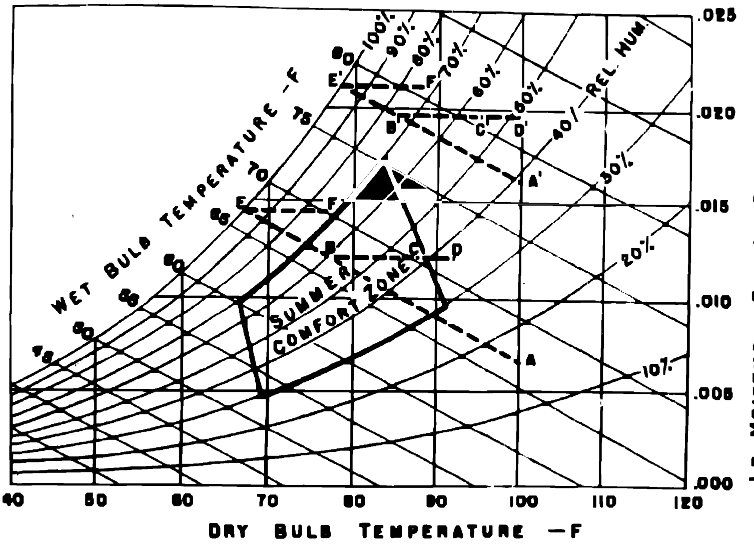


Fig. 6. Outline of Psychrometric Chart with Desert Comfort Zone

Evaporative Cooling

12. In areas where the air is very hot and dry it is possible to reduce the dry-bulb temperature by passing air over a wetted surface. The evaporation of the water requires heat; as the air touches the moisture, the heat necessary to evaporate the water is taken from the air so that the temperature of the air is lowered. However, the sensible heat is converted into latent heat. Thus total heat and the wet bulb temperature of the air remain unchanged.

If the outside air is very dry, a considerable reduction in dry bulb temperature will be obtained and at the same time the moisture content may not rise high enough to be objectionable. By studying the psychrometric chart (Fig. 6), this principle may be illustrated. Point A may be taken as outside air conditions. Point B represents a desirable air condition leaving the cooler. Point C might be a suitable air condition for discharge from the house. Point D represents an unsatisfactory condition in the last room being cooled. The maximum air cooling from conditions at A with this system is shown by point E. The line E to F then represents air travel through the house and is not satisfactory for human occupancy. Another outside air condition possible is shown by A′. The points B′, C′, D′, E′, and F′ represent the corresponding conditions that might be expected from the operation of such a cooler. No condition reached from A′ would be satisfactory.

If the air leaving an evaporative cooler is recirculated, the wet bulb temperature would continue to increase and would result in unsatisfactory conditions. There-

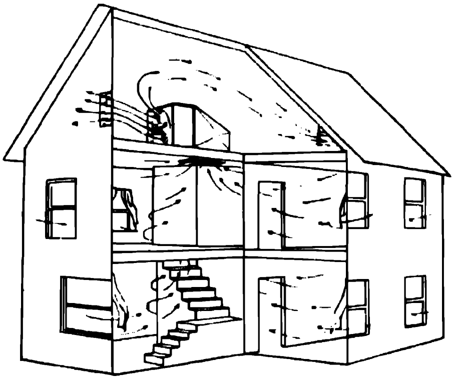


Fig. 7. Air Circulation in House with Attic Fan

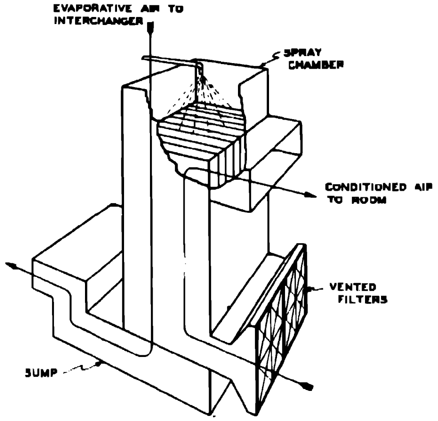


Fig. 8. Evaporative Cooler Unit, Showing Possible Air Movement

fore, fresh outside air should be used with no recirculation. This type of unit cannot be used in damp climates because the moisture content of the air leaving the unit is very high, thus raising the humidity above the comfort level.

Indirect evaporative coolers have been used to give a more positive cooling effect. This is accomplished by passing one air stream by a wetted surface to lower the dry-bulb temperature (**Fig. 8**). This air in turn is used to cool a separate air stream, with which it does not come in contact; thus the moisture content of the conditioned area remains unchanged, while the dry-bulb temperature is reduced. Since there must be a considerable difference between dry bulb and wet bulb, the indirect evaporative cooler is limited to dry climates.

Evaporative cooling is definitely limited in usefulness and cannot be called a substitute for complete air conditioning.

40. CONTROL OF AIR CONDITIONS

THIS chapter deals mainly with systems of control, but also touches upon the important related mechanical characteristics of mechanical equipment and of individual control devices. The material is taken from control problems which have been developed and solved in the field.

Load Characteristics versus Comfort Conditions

1. **Air conditioning load** includes heat gain from outside, sun heat through windows and wall surfaces, ventilation air, and various forms of internal sensible heat, such as light, power equipment, office machines and people. In addition, humidity load is introduced by ventilating air, by infiltration and by people or other internal sources. Rather wide ratios of sensible load h_s to humidity load h_L are experienced in the normal variation of these factors. In general, the h_s/h_L ratio increases with the severity of the air conditioning load, particularly so where outside weather conditions have a large influence (see Chap. 11).

Both sensible cooling and dehumidifying are accomplished at the same time by the usual **surface-type** air conditioners. Their ratio depends on psychrometric conditions which impose certain limitations. Comfort systems are usually designed to maintain temperature below a certain maximum at peak load. Comfort requirements dictate that temperature must not drop below a certain minimum. In controlling humidity as well as temperature, the control system must operate within these limits of dry-bulb temperature. For example, in an attempt to reduce relative humidity, it is not permissible to operate the equipment to cool below the limit selected, even though some air conditioning equipment will actually cause relative humidity to rise while the air conditioner operates at low dry bulb temperature.

Adjustment of Conditioning Equipment to Load

2. Reduction of the **air velocity** over the coil by changing the amount of air circulated or by bypassing some air around the coil automatically, as the cooling load decreases, reduces the sensible capacity without a corresponding decrease in moisture removal. Usually a satisfactory humidity may thus be reached before the dry bulb temperature falls to the low limit. In using the former method, distribution of air within the space must be properly maintained at the lowered velocities.

Sensible heat may be added anywhere in the system, except in the return air ducts, to provide reheat and offset the low operation required to dehumidify. **Reheat** may be by steam, electricity, hot water or condensation of the refrigerant and is normally controlled by return air temperature, on comfort installations, or from rise in humidity in cases where a higher temperature is either allowable or required.

The equipment capacity at **light load** is extremely important where a schedule of inside temperatures against outside temperatures is set up by controls. Practically, it is impossible for equipment to follow rapid outdoor temperature changes, and under certain conditions more nominal refrigeration capacity is required to maintain the inside temperature equal to the outside at, say, 75 F, than to maintain a 15 deg differential at 90 F outside.

Compressor capacity may be controlled on individual units by the use of unloading or bypass valves. Two to three steps of capacity are often employed for a single compressor. Likewise, where several compressors are employed, they may be started and stopped in any order desired to give the required steps of capacity. In many cases, multiple compressor installations may also employ bypass valves on the individual compressors. In controlling such systems, care must be taken to provide sufficiently wide steps of change in the refrigerated medium to prevent short-cycling from one step of capacity to another.

For example, in a system like **Fig. 1,** if a given step of capacity reduction is called into play at water temperature of, say 40 F, while the capacity will be restored at

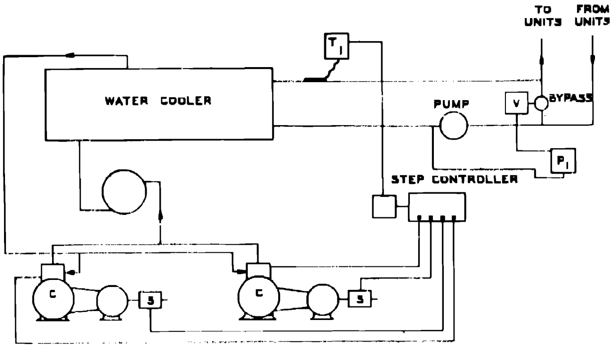


Fig. 1. Operation of Step Control of Capacity

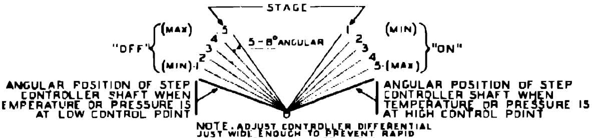

STAGE
(MAX)
"OFF"
(MIN)-1
2
3
4
5
5-8° ANGULAR
1
2
3
4
(MIN)
"ON"
5-(MAX)
ANGULAR POSITION OF STEP CONTROLLER SHAFT WHEN TEMPERATURE OR PRESSURE IS AT LOW CONTROL POINT
ANGULAR POSITION OF STEP CONTROLLER SHAFT WHEN TEMPERATURE OR PRESSURE IS AT HIGH CONTROL POINT
NOTE. ADJUST CONTROLLER DIFFERENTIAL JUST WIDE ENOUGH TO PREVENT RAPID CYCLING BETWEEN SUCCESSIVE STEPS OF CAPACITY

Fig. 1a

water temperature of 40.5 F, it is important that the reduction of refrigeration capacity shall not immediately result in a water temperature rise of 0.5 F. When step-controllers operate multispeed compressors to adjust capacity similar problems exist, but proper analysis results in an extremely flexible and satisfactory installation.

The proper positions for making and breaking of switches which operate the various capacity steps for application to this type are shown in **Fig. 1a.** Five steps of capacity are shown, but the same principles apply for any number. Note that the "on" and "off" points are grouped at the end of the step controller travel and that the differential of the controllers has been adjusted just wide enough so that short-cycling from one step to another does not occur.

If pneumatic controls are used for step operation, a similar sequence is obtained by proper adjustment of the pressure controls for each step of capacity.

Application Principles

3. In applying controls, attention to a few basic principles is important for best results. A control instrument can react only to the air immediately surrounding it. Where loads are dissimilar in various parts of the building, **separate zones** should be set up. This may be caused either from unequal exposure or unequal internal loads. For example, see **Fig. 2.** In this L-shaped building, zones 1 and 2 are never exposed to the sun, while zones 3 and 4 are exposed at different periods of the day. However, on the top floors of the building, zone 1 may have sun for a period early in the morning. This would call for at least three zones in the lower floors and four zones in the upper floors of the building. Corner rooms such as 5 may often be served partly from each adjoining zone.

In direct-expansion systems, the compressor should not be allowed to operate unless the fan of the conditioning unit is in operation. Likewise, in a water cooler, the compressor should not operate unless the circulating pump of the water cooler is operating. Proper wiring between fan or pump starter and compressor starter will provide this feature.

In addition, it is often desirable to prevent outside air from entering the conditioner, particularly in the in-between seasons when freeze-up might occur. Where

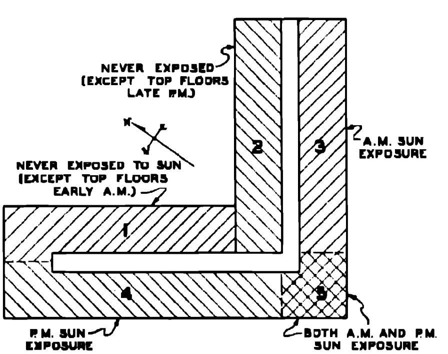

NEVER EXPOSED (EXCEPT TOP FLOORS LATE P.M.)
N
NEVER EXPOSED TO SUN (EXCEPT TOP FLOORS EARLY A.M.)
1
2
3
4
5
A.M. SUN EXPOSURE
P.M. SUN EXPOSURE
BOTH A.M. AND P.M. SUN EXPOSURE

Fig. 2. Zoning of Buildings

manual control is not satisfactory, this may be accomplished automatically with an electric damper motor which closes when the fan is off, or by a pneumatic control system using a solenoid valve in the air line to the damper motor. In either case, de-energizing the fan closes the outdoor air damper.

Sizes for **modulating valves** controlling steam or hot or cold water to coils must be chosen for the actual load, and not simply by reference to the size of existing connections. An oversize modulating valve cannot provide proper control. In modulating or floating control systems, particularly those in which several steps of compressor capacity are employed, enough change in temperature of the controlled medium must be allowed within the control range to prevent short-cycling or hunting.

Positive positioning of dampers or valves on either electrically or pneumatically operated controls is important for best operation of any system. It is now available with pneumatic systems as well as with electrically operated devices.

Where automatic changeover from **summer to winter** is to be provided under automatic control, it is important to have a dead spot between the heating and cooling functions. That is, there should be some period of inoperation in both heating and cooling between the point at which the controls will call for heating on one end of the scale or cooling on the other. Otherwise, serious cycling from one function to the other will occur.

Where refrigeration storage is employed, the proper time cycle is often established by an automatic timer. Sufficient time should be allowed in the morning to pick up the conditioning load well before any heavy load increase in the conditioned spaces.

Typical Control Systems

4. The control systems here shown are taken from actual installations and illustrate some common control methods for both large and small jobs. All figures are schematic, control functions being obtained by either electric or pneumatic devices. A single line connecting controls represents as many wires or pipes as the apparatus requires. The source of power is not shown, nor are safety controls such as high and low-pressure cutouts for the compressors.

Some of the systems shown in which controls automatically affect the compressor capacity are generally arranged on the larger installations for manual starting and stopping of the compressors, the only automatic function being capacity adjustment or emergency shutdown by the safety controls.

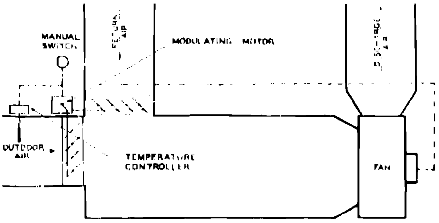


Fig. 3. Relative Amounts of Outdoor and Return Air

It is impossible to illustrate all combinations of apparatus and control, but the diagrams shown illustrate the common problems involved, and may serve as the basis for building through combinations with other apparatus to provide any of the common control sequences. For simplicity, control of the relative amounts of **outdoor and return air** are not shown on the separate figures, except when outdoor air control is tied in with other control functions. However, the system of **Fig. 3** may be used on most air conditioning systems or units. Referring to this figure, the modulating motor operates both the outdoor air and return-air dampers. The linkage is so arranged that as one damper opens, the other closes. The motor is interconnected with the fan starter so that whenever the fan is not running, the outdoor air dampers will automatically close.

When the fan runs, the outdoor temperature controller positions the motor. The settings of this temperature control are so arranged that at, say, 65 F outdoor air temperature, the maximum amount of outdoor air is introduced, and as the temperature rises above this value or falls below it, the portion of outdoor air is reduced. Proper settings of the thermostat result in a maximum of economy through the introduction of greater than the minimum amount of outdoor air whenever it can assist in cooling the space.

The minimum amount of outdoor air required for the number of people in conditioned areas can be adjusted by a manual switch so that this minimum will always be introduced except when the fan is not running. Where necessary, a second damper motor can simultaneously control exhaust dampers. This is often required in localities where larger quantities of outdoor air can be introduced economically for long periods of time.

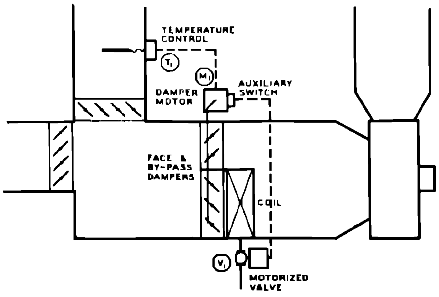


Fig. 4. Damper Control

In localities where excessively long periods of high **relative humidity** are likely to recur frequently, the motor position may also be affected by an outdoor humidistat (not shown) to prevent introducing large quantities of excessively humid air. This control arrangement depends somewhat on the flow characteristics of the dampers employed, to produce the desired mixed-air temperatures, but in general will give more uniform results than a thermostat located in the mixed air, unless the bulb of such a thermostat can be placed in the fan discharge. This is because of the difficulty of finding the proper location for the bulb in the mixed-air stream ahead of the heating or cooling equipment, so that it will be affected by the true average temperature.

It is obvious that where this or similar control systems control outdoor air in the winter, the heating and cooling coils must be of the non-freeze type.

Types of Systems

A. Control of Simple Chilled Water Unit

5. One of the simplest methods of controlling a single unit cooled by chilled water is through the use of face and bypass dampers as in **Fig. 4.** The damper motor controls these dampers so that as the bypass dampers open, the face dampers close. When wide open bypass dampers are not large enough to supply the full quantity of air to a system, it may be necessary to position the face dampers so they will not close completely in order that full quantity of air can get to the fan. An auxiliary switch on the damper motor operates to close a valve in the chilled water supply to the coil when the face dampers are completely closed and open it as soon as the face dampers start to open.

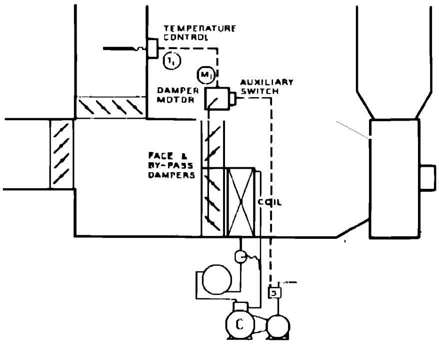


Fig. 5. Simple Control for Direct-Expansion Systems

This valve is an economy feature particularly desirable on well-water systems, where the water passing through the coils is wasted. However, it also saves considerable operating cost where units are served from a central water cooler. The damper motor itself is controlled by a temperature control in either the return-air to the unit or in the conditioned space. Usually the valve in the water supply is of the two-position type, since this results in better dehumidification under light load conditions.

Systems of this type employing modulating valves may suffer from lack of dehumidification at light loads unless the modulating range is held within the early part of the opening of the face damper.

B. Simple Control System for Direct-Expansion Unit or Central Systems

6. Fig. 5 shows a system somewhat similar to that of Fig. 4, but adapted for direct-expansion installations. As in the previous system, a return-air thermostat operates the damper motor which positions the face and bypass dampers.

Where only one or two units are cooled by a single compressor, the **linkage** to the **damper motors** should be so arranged that the minimum closure of the face dampers will permit enough air to pass over the coil to maintain a desirable suction pressure, one which will not cause serious frosting of the coil.

Frosting may be eliminated when compressors are equipped with unloading mechanisms which prevent low suction pressure, such as shown in **Fig. 14.**

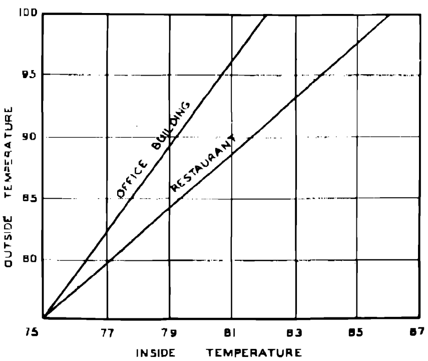


Fig. 6. Desirable Inside Temperatures

When a **single cooling unit** operates from one compressor, an auxiliary switch on the damper motor may be employed to start and stop the compressor as shown. Either the damper motor itself or the compressor motor starter should take its power through the fan starter so that the system cannot run until the fan is operating. If the system is wired in this way, the damper motor must be of the automatic closing type.

Where **several units** operate from a single compressor, the auxiliary switch from the damper motor may alternately be connected to control a valve in the liquid line to the individual unit. This arrangement is not shown, but its utility is obvious.

A **direct-expansion** arrangement with the bypass controlled in this manner ordinarily gives relatively good humidity conditions at light loads. When the cooling equipment is carefully sized to the load conditions, it is often possible to employ a rather large differential in the return-air thermostat to obtain some fluctuation in the temperature in the conditioned spaces as the load varies. In this way, on the warmer days when the load is usually highest, the temperatures will be highest, and on the cooler days when the load is least, the temperatures will be at the lower end of the range. This is desirable, since it serves to vary the differential between inside and outside, according to weather conditions.

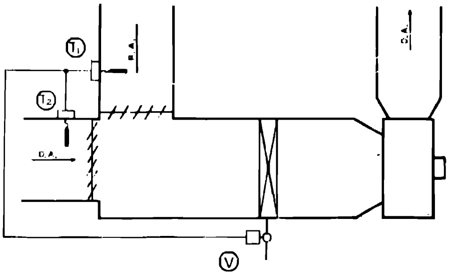


Fig. 7. Outdoor Temperature Control

C. Outdoor Compensation

7. For **short-time occupancy,** it may be desirable to vary the inside conditions in proportion to outside temperature to reduce entering and leaving sensations of the people who come into the conditioned space. This also is often done for somewhat longer periods of occupancy, such as theaters and office buildings, but the variation is usually not so great in these cases.

Fig. 6 shows desirable inside temperatures for use with various outside temperatures, found to produce reasonable degrees of comfort in the northern and central zones of the United States. Such a schedule may be set up automatically by means of a control system such as that in **Fig. 7**.

The **outdoor temperature controller** measures the outside temperature and determines the point at which the inside temperature controller will call for cooling if a two-position control is employed. If modulating control is used, the modulating range through which the valve will be positioned by the inside controller is varied. The return-air thermostat of Fig. 7 operates a valve to control the amount of cooling fluid admitted to the coil. Alternately, it could of course start and stop the compressor of a direct-expansion system.

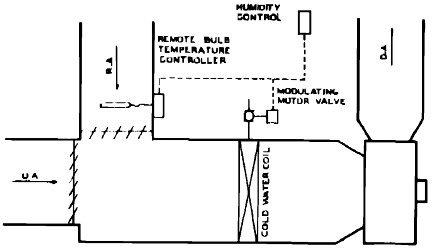


Fig. 8. Motorized Valve Control

As previously mentioned, inside temperatures cannot be expected to follow outdoor temperature fluctuations rapidly. Many engineers do not consider it desirable that they should. However, this type of control system will follow day-to-day trends in outdoor temperature, and has the advantage that when outdoor temperature is low, a lower inside temperature is held, which normally results in better relative humidity conditions.

D. Modulating Control with High Relative Humidity during Light Loads

8. The advantages of modulating control of the cooling fluid are fairly obvious, since rapid fluctuations in delivered air temperatures are eliminated, resulting in less disturbance to air distribution within the conditioned spaces and contributing to improved comfort. But where chilled water or brine is used as the cooling medium, the average temperature of the water in the coil is raised as its flow is throttled. This results in a smaller proportion of dehumidification, and on jobs where high humidities may be troublesome, uncomfortable conditions may be encountered, particularly at light loads.

To overcome this, the system of **Fig. 8** may be employed. The return-air temperature controller ordinarily operates to modulate the motorized valve to control the flow of cold water through the cooling coil. However, if the humidity rises to a predetermined high value, the humidistat causes the remote-bulb temperature controller to operate the valve in a two-position manner and thus provides a maximum of dehumidification during the cooling period, and yet the system is prevented from cooling to too low a temperature by the temperature controller.

In this way, the advantages of modulating control are maintained as long as practical, and the system reverts to two-position control whenever greater dehumidification is required. This system may also be used with direct-expansion coils by means of modulating attachments to the expansion valves or modulating liquid valves, but it should be noted that on most such systems, it is not necessary to revert to two-position operation, since the suction pressure automatically reduces as refrigerant is throttled, due to the re-established balance between coil capacity and machine capacity, which provides a greater ratio of dehumidification. The amount of reduction in suction pressure which is permissible is

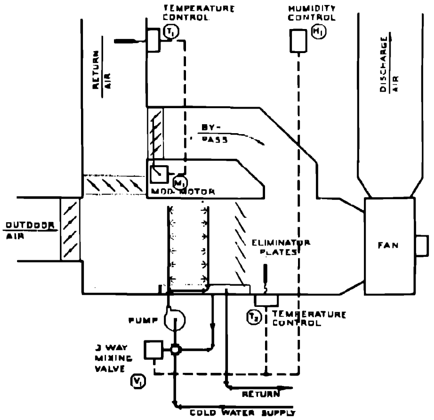


Fig. 9. Control of Air Washer Cycle

limited by the lowest practical coil temperature, and hence, even on direct-expansion systems, it is sometimes desirable to revert to two-position operation when humidity is high.

E. Air-Washer Control

9. Many medium and larger-sized systems employ air washers instead of extended surfaces for cooling. The degree of automatic control employed with air-washer equipment has increased in recent years, and **Fig. 9** shows a fairly complete control of the cooling cycle of an air-washer system. The outdoor air control shown in Fig. 3 could be applied very nicely to an air washer of this type, particularly during the cooling cycle. Cold water could be supplied from a system such as that shown in Fig. 1. There are times when a face damper located completely across the face of the air washer is necessary to obtain proper air pressure relationships so that the amount of air bypassing can be controlled.

The pump sprays water into the air stream, from which it is collected in the pan and returned to the pump suction. A three-way mixing valve is controlled by dew-point temperature controller T_2. This valve mixes the cold supply water and the water from the pan to give a constant dew point to the air leaving the eliminator plates. This occurs since the air is practically saturated in passing through the air washer and hence the temperature of the air equals its dew point.

The temperature within the conditioned space is controlled by return-air and bypass dampers operated by the damper motor M_1, controlled by the return-air temperature controller T_1.

With this much control, the relative humidity attained in the space will vary somewhat, since smaller quantities of air pass through the sprays and are treated to the same dew point temperature at light loads. Where it is necessary to overcome this to give a more constant relative humidity, the h_s/h_L ratio can be varied considerably by compensating the action of the dew point controller through a humidity control H_1, located in the conditioned space. This control measures the relative humidity in the space and determines the dew point to be maintained by T_2. The higher the relative humidity, the lower the dew point produced by the combined action of the three-way mixing valve and temperature controller T_2. Also, T_2 and H_1 could control a straightway valve in the cold water supply line with a check valve in the pump suction from the pan, and obtain the same results as with a three-way valve.

With this type of control, the lowest h_s/h_L ratio attainable is determined by the lowest cold-water supply temperature obtainable.

F. Blower Type Zone Installation

10. **Fig. 10** shows a system suitable for some of the larger zone jobs employing a central blow-through type fan. With this arrangement it is usually practical to control the outdoor air and return-air dampers from the mixed-air temperature so that when no cooling or heating is required, the proportion of the two will be such as to give 65 F air whenever such a mixture is possible. The minimum outdoor air quantity would ordinarily be set by a manual switch and the outdoor air damper motor should be so connected that when the fan is stopped, the outdoor air dampers are closed tightly.

The temperature of the air leaving the fan will be at, say, 65 F, whenever outdoor air is sufficiently cool to produce this low a temperature in the mixture. Other-

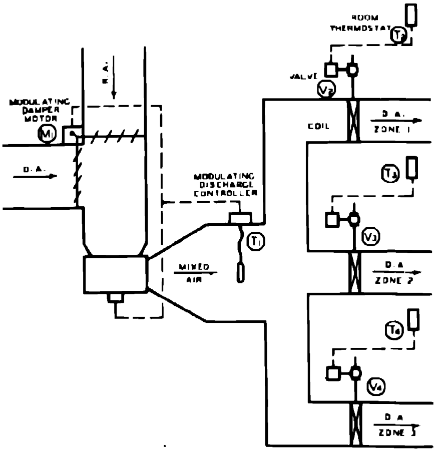


Fig. 10. Blower Type Zone Installation

wise, it will be at some higher value. This air is delivered through a duct system to the separate zones 1, 2 and 3, in each of which is located a cooling coil. The cooling provided by each coil is controlled by a room thermostat located in the corresponding zone.

Such a system might be very applicable to a building such as that shown in Fig. 2. The separate zones may be controlled either from modulating or two-position room thermostats, and individual coils may be equipped with bypass around the coils wherever desired. When so used, satisfactory temperature and relative humidity conditions should be experienced over normal ranges of load.

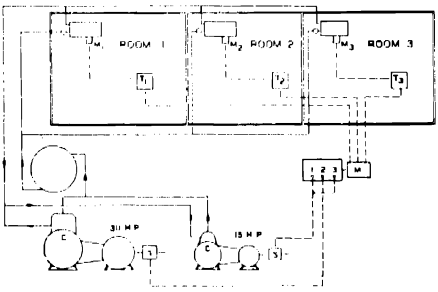


Fig. 11. Compressor Capacity Control

This system is also used for heating where zoning is necessary, by having the individual room thermostats control modulating steam valves on individual heating coils in the various ducts.

A common variation of the system shown on Fig. 10 is to use a fan discharging from a central duct system to individual zones where booster fans, individual air heaters and supply and bypass dampers are employed. This arrangement helps to reduce duct sizes in buildings.

On some installations, both cooling and heating coils are employed, so that the same system performs the dual function of heating and cooling. From the control point of view, it is better to have the air flow first through the cooling coil and then through the heating coil unless other considerations make this arrangement impractical. When this is done, it becomes entirely practical to heat in some zones while cooling in others or to cool whenever it is necessary to keep the humidity low in a given zone and to provide reheat in the heating coil of that zone to prevent excessive lowering of temperatures. Reheat could also be supplied by utilizing the waste heat of the condenser gases or the heat in the condenser water.

G. Control of Compressor Capacity for Zone Systems

11. Various methods are employed with zone systems such as that shown in Fig. 10 to vary compressor capacity. Compressor bypass valves, clearance pockets or two-speed motors may be automatically controlled to vary capacity, or a number of units of different horsepower may be used to provide various steps of capacity under automatic control. **Fig. 11** shows a system of the latter type in which one compressor is twice the capacity of the other. By turning on the smaller, then starting the larger and stopping the smaller, and then adding the smaller compressor, three steps of capacity are possible with only two compressors.

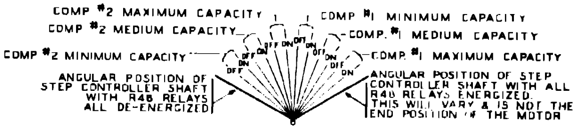
A step-controller is ordinarily employed to produce this sequence of capacity. The step-controller may be operated in response to suction pressure where conditions permit. However, there is less danger of short-cycling any of the steps of capacity if they can be cut in and out from room conditions.

The system of Fig. 11 accomplishes this as follows. The room thermostats in the separate zones have a cumulative effect on the step-controller M so that the greater the demand for cooling, the greater will be the compressor capacity employed. At the same time, these thermostats in individual zones can control motors M on face and bypass dampers, which in turn can close valves in liquid lines to the individual cooling units if required.

This control system assures that the refrigeration capacity shall approximately match the load conditions. The position of the step-controller represents the sum of the loads in the various zones, and since these loads change rather slowly, the danger of short-cycling from one step of capacity to another is largely eliminated. It is well to protect a system of this type from low suction pressure by a suction pressure switch, independent of the other controls.

The adjustment of the step controllers for applications of this type in which the summation of the total load is measured, should be as shown in **Fig. 12**. Note that this differs materially from the application discussed relative to Fig. 1.

Fig. 12


COMP #2 MAXIMUM CAPACITY
COMP #2 MEDIUM CAPACITY
COMP #2 MINIMUM CAPACITY
ANGULAR POSITION OF STEP CONTROLLER SHAFT WITH R4B RELAYS ALL DE-ENERGIZED
COMP #1 MINIMUM CAPACITY
COMP. #1 MEDIUM CAPACITY
COMP. #1 MAXIMUM CAPACITY
ANGULAR POSITION OF STEP CONTROLLER SHAFT WITH ALL R4B RELAYS ENERGIZED. THIS WILL VARY & IS NOT THE END POSITION OF THE MOTOR
OFF
ON

For pneumatic controls, the same operating sequence should be followed.

This method of controlling the step-controller is just as applicable when controlling compressor bypass valves or a multi-speed motor on a single compressor. If one zone has a greater load than another, the effect of its associated thermostat on the step-controller can be correspondingly greater than that of other zones.

Many compressors are now equipped with capacity control which unloads the individual cylinders successively by holding open their suction valves. This action is usually responsive to suction pressure, and thus the adaptation of compressor capacity to load is automatic. A typical mechanism of this type is shown in Fig. 14. An outstanding advantage of this method is that the mechanism can be allowed to respond to very small changes in suction pressure and thus any desired pressure can be accurately obtained whether the compressor has one, or ten, or more cylinders.

The dotted lines of Fig. 14 show the addition of **pneumatic control** equipment to vary the operating suction pressure in response to temperature or humidity as measured in the conditioned spaces, or wherever else desired. Similar means for adjusting the suction pressure with electrical controls may be used.

H. All-Year System

12. **Fig. 13** shows a typical all-year system. Outdoor air and return air are proportioned according to the demand of temperature controllers T_2 and T_3, so that when outdoor temperature is either above or below a mean value, the amount of outdoor air is decreased until the minimum set by S_1 is reached.

The cooling cycle is controlled by T_1 and H_1 in the return air. The face and bypass dampers of the coil are affected only by the temperature of the return air, so that a definite low limit for the cooling

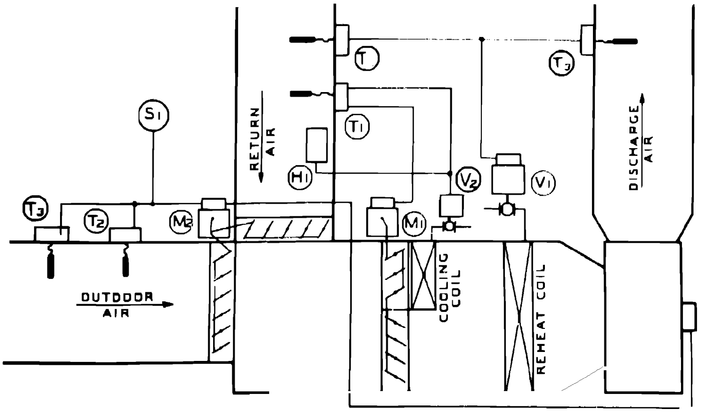


Fig. 13. Typical All-Year System

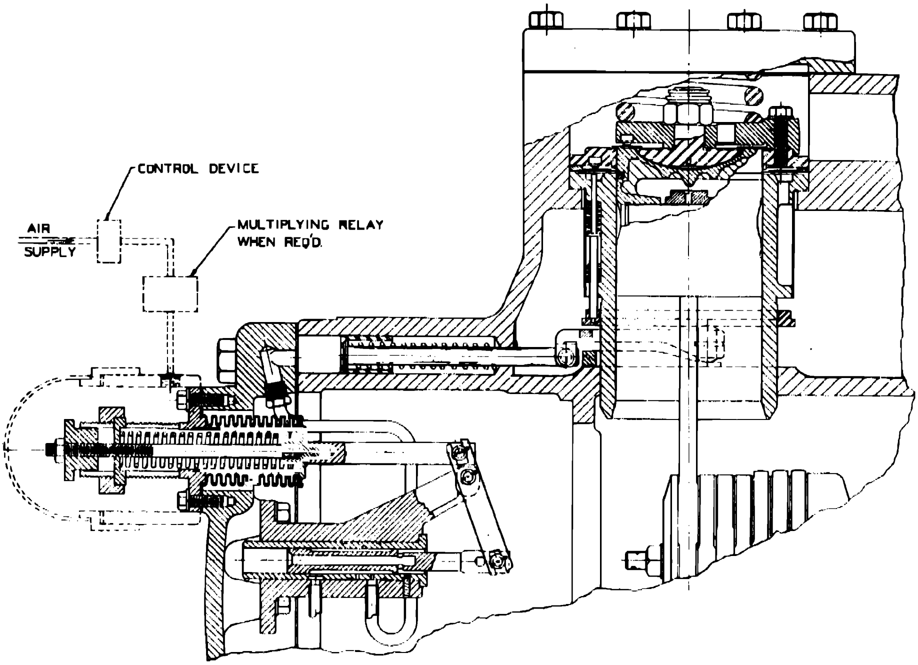


Fig. 14

cycle is provided. As the temperature nears its low limit, the cooling fluid to the coil is throttled for economy by T_1.

At any time when humidity is high, the humidistat H_1 keeps valve V_2 open to provide a greater ratio of dehumidification. If reheat is available all year, V_2 may be connected to M_1 to open the bypass dampers at least to a minimum position whenever V_2 opens wide, as by humidistat action. Further reduction in temperature causes return-air controller T_5 to begin to open the heating valve V_1 and provide the necessary heat to maintain constant temperatures. Usually a discharge temperature control T_4 is required in the winter time to prevent the discharge of air at an uncomfortably low temperature.

In the **cooling cycle,** this thermostat will sometimes need to be made inoperative, which can be done by a manual switch or by extra contacts in T which operate on a temperature rise somewhat above the point at which the heating valve closes.

This system, with the proper settings of T_1 and T, can be made to give automatic changeover from heating to cooling, but is also adaptable to manual changeover by the addition of the proper manual switches.

Winter humidity control can be supplied by the addition of humidifying sprays, pan-type humidifiers or other humidifying equipment, controlled by a return-air or space humidistat. Humidifying equipment, like the outdoor air motor, should take its power from the fan power supply so that it cannot operate except when the fan is running.

I. Refrigeration Storage

13. Since many air conditioning systems have rather short peak loads either from weather or occupancy, smaller refrigeration equipment can adequately condition such spaces if storage for refrigerating effect is provided and the smaller equipment allowed to run at least a portion of the time while cooling is not actually required.

Two types of systems are used—one in which a **water cooler,** such as shown in Fig. 1, is used to chill water and store it at above freezing temperature in a large insulated storage tank. In this case, a control system such as shown in Fig. 1 can be used for the control of the refrigeration equipment. The thermostat in the water supply line to the units and storage tank is ordinarily set somewhat lower than would otherwise be the case. Control of the individual units can be accomplished from any of the systems applicable which are mentioned above.

In the second system, a special type of water cooler is employed, in which ice may be accumulated without danger of bursting tubing or otherwise damaging the cooler. In this case, control of the compressor is accomplished either from a remote-bulb temperature controller with the bulb so located with respect to the evaporator in the water cooler as to limit the thickness of ice formation, or it may be controlled from a timed cycle so that during a certain period of time, say at night, the refrigeration compressor will run continuously to build up ice.

PART VII

MISCELLANEOUS

Chapter *Page*

41. Glossary of Refrigeration—B. H. Jennings 657
The definitions of this glossary have been freely drawn and cross checked from many authoritative sources, and it is hoped the glossary represents usages accepted by the refrigeration industry. In preparing this glossary, the editor has received invaluable help from many individuals with the suggestions of Messrs. Glenn Muffly, F. H. Faust, Gayle B. Priester, and L. C. Bastian being most extensive.

42. Codes and Standards 677

41. GLOSSARY OF REFRIGERATION

absolute zero—the zero point on the absolute temperature scale, 459.7 degrees below the zero of the Fahrenheit scale, 273.16 degrees below zero on the Centrigrade scale.

absorbent—a material which has the ability to take up water vapor or some other vapor but which changes chemically, physically or both, during the cycle. Calcium chloride is an example of a solid-liquid material, while liquid materials include solutions of lithium chloride, lithium bromide and the ethylene glycols.

absorber—a device containing liquid for absorbing refrigerant vapor or other vapors. In an absorption system that part of the low side used for absorbing refrigerant vapor.

absorption—a process whereby a liquid substance can act in such manner as to receive another liquid or gaseous fluid which in turn becomes part of the common resulting solution.

absorption system—a refrigeration system in which the refrigerant gas evolved in the evaporator is taken up in an absorber and released in a generator upon the application of heat.

acceleration due to gravity—the rate of increase in velocity of a freely falling body ignoring the buoyant effects of the air. Its value varies with latitude and elevation. The International Standard taken at sea level and 45 deg latitude is 980.665 cm per second per second or 32.174 ft per second per second, symbol (*g*).

access doors—doors provided in a unit to permit inspection of the inside, as of plenum chambers.

accumulator—a storage chamber for low-side liquid refrigerant.

activated alumina—a form of aluminum oxide which absorbs moisture readily and is used as a drying agent.

adiabatic process—a thermodynamic process during which no heat is extracted from or added to the system.

adsorbent—a material which has the ability to cause molecules of gases, liquids or solids to adhere to its internal surfaces without changing the adsorbent physically or chemically. Certain solid materials such as silica gel, activated carbon and activated alumina have this property.

adsorption—the adherence of molecules of dissolved substances, gases or liquids to the surfaces with which they are in contact.

aeration—a term generally employed with reference to air circulation or ventilation. In milk cooling a method where the milk flows over refrigerated surfaces exposed to the atmosphere. In ice manufacturing a misnomer for agitation by means of air.

agitator—a device causing turbulent motion in a fluid confined in a tank.

air-agitation system—a combination consisting of a power-driven blower, distributing piping and flexibly connected fittings for delivering air to the water in ice cans for the purpose of agitating the water and promoting the production of clear ice.

air alternator—a device which automatically switches the air from one side of the ice tank to the other.

air blast—forced air circulation.

air changes—the amount of air leakage is sometimes computed by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room.

air circulation—natural or imparted motion of air.

air cleaner—a device used to remove airborne impurities (see air washers and air filters).

air conditioning—the simultaneous control of all or at least the first three of the following factors affecting both the physical and chemical conditions of the atmosphere within a structure; temperature, humidity, motion, distribution, pressure, dust, bacteria, odors, toxic gases and ionization.

air conditioning unit—an assembly of equipment for the control of at least the first three items enumerated in the definition of *air conditioning*.

air cooler, forced-circulation—a factory-made encased assembly of elements by which heat is transferred from air to evaporating refrigerant.

air laterals—pipe which supplies the air to the ice cans contained in the freezing tank.

air mains—pipe which carries air to the air laterals supplying ice cans contained in the freezing tank.

air, saturated—a mixture of dry air and saturated water vapor, both at the same dry-bulb temperature.

air tunnel—a refrigerated tunnel with rapid air circulation through which the product to be frozen is passed.

air washer—a water-spray system or device for cleaning, humidifying or dehumidifying the air.

air washer and cooler—a vessel designed for passing agitating air through a body of chilled water for the purpose of cleaning and dehumidifying the air delivered to ice cans for agitation of the water therein.

ambient air—generally speaking, the air surrounding an object. In a domestic or commercial refrigerating system having an air-cooled condenser, the temperature of the air entering the condenser.

ammonia machine—shortened expression for a compression refrigerating machine using ammonia as a refrigerant. Similarly, sulfur-dioxide machine, etc.

analyzer—device in high side of absorption system for increasing the concentration of refrigerant in the vapor entering rectifier or condenser.

anemometer—instrument for measuring the velocity of air or other gas.

anticipating control—one which is artificially forced to cut in before it otherwise would, thus starting the cooling (or heating) before needed, to reduce the temperature fluctuation.

apparatus dew point—the dew point temperature of the air leaving the conditioner unit (i.e., humidifier, dehumidifier, or surface cooler).

aspiration—production of movement in a fluid by suction.

atomize—reduce to fine spray.

attic fan—an exhaust fan to discharge air near the top of a building while cooler air is forced (drawn) in at a lower level.

available energy—the portion of the total energy which can be converted to work in a perfect engine.

back pressure—loose terminology suction pressure of refrigeration vapor in a system.

baffle—partition used to divert fluid flow.

balling—a name applied to the hydrometric scale used in measuring the strength of worts. This scale gives directly the percentage extract dissolved in the wort.

barometer—instrument for measuring atmospheric pressure.

benching machine—portable device for elevating and stacking ice blocks.

bimetallic element—one formed of 2 metals having different coefficients of thermal expansion, used as a temperature control device.

binary vapor cycle—a refrigerating cycle in which two separate refrigerants are used, one superimposed upon and augmenting the cycle of the other.

blanch—scald or sterilize, in case of vegetables to be canned, cooked or frozen, usually by dipping in a hot-water bath.

bleeder—pipe attached, as to a condenser, to lead off liquid refrigerant, parallel to main flow.

blower—enclosed fan device or low-pressure compressor used to impart motion to air.

boiler—a closed vessel in which steam is generated or in which water is heated, also a refrigerant evaporator.

boiling point—the temperature at which a liquid vaporizes upon the addition of heat, dependent upon the absolute pressure at the liquid-vapor surface.

booster—a compressor for very low pressures, usually discharging into the suction line of another compressor.

bore—inside diameter of a cylinder.

brazed—joined by fusion using a spelter on the order of brass. This is considered equivalent to hard soldering.

breaker—a relatively poor conductor of heat used to join the liner and outer shell of an internally refrigerated container such as a refrigerator. A frame making a thermal break around a door or its opening to retard heat flow to the interior of a refrigerator.

brine—any liquid cooled by a refrigerating system and used for the transmission of heat.

brine cooler—evaporator for cooling brine in an indirect system.

brine-spray system—refrigerating scheme for cooling by a mist or spray of brine.

brine tank—in an ice plant, the main freezing tank in which the cans are immersed while ice is being produced. In a brine circulating system, a storage tank or balance tank for brine. In domestic and commercial fields, a container surrounding the evaporator and filled with brine for storing refrigeration or for equalizing temperature at various points of the evaporator, especially in ice cream cabinets.

British thermal unit (Btu)—heat required to produce a temperature rise of 1 degree Fahrenheit in 1 lb of water. Strictly, 1 Btu =251.996 I.T. calories, where the I.T. calorie is 1/860 kilowatt hour. The mean Btu is 1/180 of the energy required to heat water from 32 deg F to 212 deg F.

bunker—space in refrigerator given to ice or cooling element.

bypass—a pipe or *duct*, usually controlled by valve or *damper*, for conveying a fluid around an element of a system.

"C" factor—thermal conductance—the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of nonhomogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plasterboard, hollow clay tile, etc.

cabinet—refrigerating container per se; mechanical refrigerator without refrigerating machine.

calibration—process of dividing and numbering the scale of an instrument; also of correcting or determining the error of an existing scale, or of evaluating one quantity in terms of readings of another.

calorie—heat required to raise the temperature of 1 gram of water 1 degree centigrade, actually from 4C to 5C. Mean calorie =1/100 part of the heat required to raise 1 gram of water from 0 to 100 degrees centigrade. Great calorie or kilo calorie =1,000 small calories.

calorimeter—device for measuring heat quantities, such as machine capacity, heat of combustion, specific heat, vital heat, heat leakage, etc.; also device for measuring quality (or moisture content) of steam or other vapor.

can dump—a device so arranged that ice cans conveyed from the tank may be placed therein approximately vertically, then tipped in order that the ice block may slide out. It may be equipped with sprinkling pipes for spraying the can and thawing the ice block free.

capacitor—a condenser type of motor, or the electrical condenser connected with it.

capacity—of a refrigerating compressor is the refrigerating effect in Btu per hour produced by the change in total enthalpy between the refrigerant liquid at a temperature corresponding to the pressure of the vapor leaving the compressor and the total enthalpy of the refrigerant vapor entering the compressor per hour. Of a condensing unit, the refrigerating effect in Btu per hour produced by the change in total enthalpy of refrigerant liquid leaving the unit and the total enthalpy of the refrigerant vapor entering the unit. Generally measured in tons per day or Btu per hour.

capacity, heat—the amount of heat necessary to raise the temperature of a body one degree. Numerically the mass multiplied by the specific heat.

capacity reducer—in a compressor, a device such as a clearance pocket, movable cylinder head, or suction bypass, by which compressor capacity can be adjusted without otherwise changing the operating conditions.

capillarity—the action by which the surface of a liquid where it is in contact with a solid (as in a slender tube) is raised or lowered.

capillary tube—a vapor lock device comprising a tube of small ID; also a small diameter tube connecting a thermostatic bulb with the bellows or diaphragm of a control device.

carbon dioxide ice—solid CO_2, dry ice.

carbonization—carbon deposits produced by oxidation of the lubricating oil, which may seriously interfere with the proper functioning of the compressor valves.

Carnot cycle—a sequence of operations forming the reversible working cycle of an ideal heat engine of maximum thermal efficiency. It consists of isothermal expansion, adiabatic expansion, isothermal compression and adiabatic compression to the initial state

cascade system—one having two or more refrigerant circuits, each with a pressure-imposing element, condenser and evaporator, where the evaporator of one circuit cools the condenser of the other (lower-temperature) circuit.

carrying freezer—cold storage freeze room, generally kept between −20 F and 20 F, to receive and hold frozen goods.

centigrade—thermometric system in which the freezing point of water is called 0° and its boiling point 100° at normal atmospheric pressure (14.696 psi).

central fan system—a mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and conveyed to and from the rooms by means of a fan and a system of distributing ducts.

central plant system—a system with two or more low sides connected to a single, central high side, a multiple system.

centifugal compressor—a nonpositive displacement *compressor* which depends for pressure rise, at least in part, on centrifugal effect.

centrifuge—device for separating substances of different densities by centrifugal action.

change of air—introduction of new, cleansed or recirculated air to conditioned space, measured by the number of complete changes per unit time.

change of state—change from one phase, such as solid, liquid, or gas, to another.

charge—amount of refrigerant in a system; charging, putting in charge.

chill—to apply refrigeration moderately, as to meats, without freezing.

chilling room—room where animal carcasses are cooled after dressing prior to cold storage.

chimney effect—the tendency in a duct or other vertical air passage for air to rise when heated owing to its decrease in density.

class of refrigerating system—formerly in extensive use, now becoming obsolete as a result of code change to classification rather than weight; refers to total weight of refrigerant contained; Class A system is one containing 1,000 lb or more of refrigerant. Class B system is one containing more than 100 lb but less than 1,000 lb of refrigerant. Class C system is one containing more than 20 lb but not more than 100 lb of refrigerant. Class D system is one containing more than 6 lb but not more than 20 lb of refrigerant. Class E system is one containing 6 lb or less of refrigerant.

clearance—space in cylinder not occupied by piston at end of compression stroke, or volume of gas remaining in cylinder at same point. Measured in percentage of piston displacement.

clearance pocket—in a compressor, a space of controlled volume to give the effect of greater or less cylinder clearance, thereby changing compressor capacity.

closed cycle—any cycle in which the primary medium is always enclosed and repeats the same sequence of events.

coefficient of expansion—numerical ratio of the increase in length or volume per degree rise in temperature.

coefficient of heat transmission—the amount of heat (Btu) transmitted from fluid to fluid in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

coefficient of performance—ratio of the refrigeration produced to the work supplied, refrigeration and work each expressed in the same thermal units, i.e., Btu/Btu; cal/cal.

coil—a cooling or heating element made of pipe or tubing.

cold storage—a trade or process of preserving perishables on a large scale by refrigeration.

comfort air conditioning—the simultaneous control of all, or at least the first three, of the following factors affecting the physical and chemical conditions of the atmosphere within a structure for the purpose of human comfort; temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases and ionization, most of which affect in greater or lesser degree human health or comfort.

comfort chart—a chart showing effective temperature and humidities under which human comfort is probable; usually drawn on a psychrometric chart base.

comfort cooling—refrigeration for comfort as opposed to refrigeration for storage or manufacture.

comfort line—the effective temperature at which the largest percentage of adults feel comfortable.

comfort zone—(average) the range of effective temperatures over which the majority (50% or more) of adults feel comfortable. (Extreme) The range of effective temperatures over which one or more adults feel comfortable.

commercial refrigerator—reach-in or service refrigerator of commercial size with or without means of refrigeration.

commercial system—refrigeration system used in a commercial and/or a business place, such as a meat market, store, florist shop, hotel, office building, restaurant, candy shop, bakery, or other places of similar commercial enterprise, assembled and installed in the manufacturing and/or business portion of any building (code).

compression, compound—compression by stages in two or more cylinders.

compression, single-stage—compression in one stage.

compression system—refrigerating system in which the pressure imposing element is mechanically operated.

compressor, compound—a compressor in which compression is accomplished by stages, as in two or more cylinders.

compressor, double-acting—one which has two compression strokes per revolution of the crank shaft per cylinder, i.e., both faces of the piston are working faces.

compressor, refrigerant—a specific machine for compressing a given refrigerant vapor from an evaporating system and discharging it into a condensing system (code). (2) That part of a mechanical refrigerating system which receives the refrigerant vapor at low pressure and compresses it into a lower volume at higher pressure.

compressor, single-acting—one having one compression stroke per revolution of the crank for each cylinder.

compressor unit—a condensing unit less the condenser and liquid receiver.

concentration—a number specifying the composition of a solution with respect to the constituent named, as pounds of salt per gallon of brine.

condenser, atmospheric—condenser operated with water which is exposed to the atmosphere.

condenser (liquefier)—a vessel or arrangement of pipe or tubing in which the vaporized (and compressed) refrigerant is liquefied by removal of heat.

condenser, water-cooled refrigerant—an assembly of elements by which the flows of refrigerant vapor and water are maintained in such a heat transfer relationship that the refrigerant vapor is condensed into a liquid and the water is heated.

condensing refrigeration effect—condensing heat rejection effect, less the heat added to the refrigerant vapor in the refrigerant compressor.

condensing unit—a specific refrigerating machine combination for a given refrigerant, consisting of one or more power-driven compressors, condensers, liquid receivers (when required), and the regularly-furnished accessories.

conductance, thermal—of a body per unit area between two isothermal surfaces is the rate of heat flow through the body per degree difference between the temperatures of the isothermal surfaces.

conduction—heat transmission, as by particle to particle in a homogeneous substance, or by contact of two substances.

conductor (heat)—a material capable of readily conducting heat. The opposite of an insulator or insulation.

connecting rod—a device connecting the piston to a crank and used to change rotating motion into reciprocating motion, or vice versa, as from rotating crankshaft to reciprocating piston.

conservation of energy—see first law of thermodynamics.

control—any device for regulation of a machine in normal operation, manual or automatic. If automatic, the implication is that it is responsive to changes of temperature, pressure, liquid level or with time.

controlled atmosphere storage—(gas storage) artificial addition of carbon dioxide to the atmosphere, particularly in large concentration, with no attempt to regulate the amount of oxygen.

convection—transfer of heat by movement of fluid containing thermal energy.

convection, natural—circulation of gas or liquid (usually air or water) due to its own thermal expansion; stack effect. In water there is a zone of reverse flow due to thermal contraction as water is warmed from 32 to 39.2 degrees fahrenheit, the maximum density point of water being 4 degrees centrigrade or 39.2 degrees fahrenheit.

convector—a heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection. Such a surface may or may not be enclosed or concealed. When concealed and enclosed, the resulting device is sometimes referred to as a concealed radiator.

cooler—in cold storage practice, an insulated room maintained at 30 F or above.

or in commercial systems, a mechanical device (i.e., not a cool space or box), as *unit cooler* or *room cooler*, usually called a cooling element.

or cooling unit—specific air treating combination consisting of means for air circulation and cooling.

cooling (or heating) air conditioning unit—a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer, with control means for cooling (or heating) and maintaining humidity.

cooling of air—reduction in air temperature due to the abstraction of sensible heat as a result of contact with a medium held at a temperature lower than that of the air. Cooling may be accompanied by moisture addition (evaporation), by moisture extraction (dehumidification) or by no change whatever of moisture content. Moisture change, if present, is considered a secondary or by-product effect. The medium may be directly in contact with the air (as water, brine, or ice) or indirectly through a barrier wall (as cooling surface). When the latter method is used, and the surface temperature is held above the air dew point, only cooling occurs without moisture inter-change.

cooling element—heat transfer surface containing refrigerating fluid in location where refrigerating effect is desired.

cooling tower—device for lowering the temperature of water by evaporative cooling, in which the water is showered through a space through which outside air circulates.

cooling water—water used for condensation of refrigerant; condenser water.

copper plating—a film of copper deposited by electrical, immersion or other means on the surface of another material such as iron or steel. In refrigeration usually on compressor walls, pistons, discharge valves and shaft or seal.

core area—the total plane area of the portion of a grill, face, or register, bounded by a line tangent to the outer edges of the outer openings through which air can pass.

core sucking and filling system—a combination consisting of a pump, distributing piping, hose, and device by which water containing impurities may be withdrawn from the unfrozen core of an ice block, and by means of connection to the source of water supply the core may be refilled with pure water.

corkboard (insulation)—cork granules, cleaned, compressed and bonded by heat to a weight of about 0.65 lb per board foot.

corresponding values—simultaneous values of various properties of a fluid, such as pressure, volume, temperature, etc., for a given condition of fluid.

corrosive—having rusting or chemically destructive effect on metals (occasionally on other materials).

counterflow—in heat exchange between two fluids, opposite direction of flow, coldest portion of one meeting coldest portion of the other.

critical point—of a substance, state point at which liquid and vapor have identical properties; critical temperature, pressure and volume are those properties at the critical point. Above the critical temperature or pressure there is no line of demarcation between liquid and gaseous phases.

cryohydrate—a frozen mixture of water and a salt; a brine mixed in eutectic proportions to give the lowest freezing point.

crystal formation, zone of maximum—temperature range in freezing in which most of the freezing takes place (about 25 to 30 F for water).

cutting room (locker plants)—cold room where animal carcasses are cut up into commercial sizes such as rib roast, legs, etc.

cycle—complete course of operation of refrigerant back to a starting point, measured in thermodynamic terms (functions); also used in general for any repeated process or any system.

Dalton's Law of partial pressure—each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

damper—a valve, or plate, used to regulate the flow of air or other fluid.

damper (multiple louvre)—a damper having a number of adjustable blades.

decibel—a unit commonly used for expressing sound or noise intensities referred to an arbitrary reference level. It is defined as *db difference* $= 10 \log_{10}(P_1/P_0)$, where P_1 is the unknown intensity. P_0 the reference level is commonly taken as 10^{-16} watts per sq centimeter which corresponds to the threshold of hearing at 1,000 cycles per second.

deck, coil deck—insulated horizontal partition between refrigerated space and bunker.

decomposition—process of chemical change, breaking up of structures, spoilage.

defrosting—removal of accumulated ice from cooling element, thawing frozen foods.

defrosting cycle—a refrigeration cycle which permits cooling unit to defrost during off period.

degree-day—degrees × days; a unit, based upon temperature differences and time, used in specifying the nominal heating load in winter. For any one day there exist as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over 24-hour period, and a temperature of 65 F.

dehumidification—the condensation of water vapor from air by cooling below the dew point or removal of water vapor from air by chemical or physical methods.

dehumidifier—an air cooler or washer used for lowering the moisture content of the air passing through it; an absorption or adsorption device for removing moisture from air.

dehumidifier, surface—an air conditioning unit, designed primarily for cooling and dehumidifying air through the action of passing the air over wet cooling coils.

dehumidifying effect—of a unit in Btu per hr; the difference between the moisture content, in pounds per hour, of the entering and leaving air multiplied by 1,060.

dehydration—the removal of water vapor from air by the use of absorbing or adsorbing materials. (2) The removal of water from stored goods.

dehydrator—a device used to remove water or water vapor from refrigerant.

density—mass or weight per unit of volume.

dense-air system—cold-air system maintained under pressure greater than atmospheric in which air is compressed, heat of compression dissipated, and the air, chilled by expansion and performance of work, can create useful refrigeration.

desiccation—drying by evaporation, generally due to low humidity.

design working pressure—the maximum allowable working pressure for which a specific part of a system is designed.

dew point—temperature at which condensation starts if moist air is cooled at constant pressure with no loss or gain of moisture during the cooling process.

dew point rise—increase in moisture content (specific humidity) of air expressed in terms of rise in dew point temperature.

diesel engine—a power plant used to change thermal energy to mechanical energy using the heat of compression to ignite the fuel.

differential (of a control)—the difference between cut-in and cut-out temperatures or pressures.

diffusers, air—see grilles.

diffusion temperature difference—the temperature difference between the air temperature at supply opening and design indoor temperature.

dip tanks—a tank located convenient to the can dump and supplied with water in which the ice can is immersed to thaw the ice block loose from the can.

direct connected—driver and driven, as motor and compressor, positively connected in line to operate at same speed.

direct expansion coils—coils using the direct method of refrigeration.

direct method of refrigeration—a system in which the evaporator is in direct contact with the material or space refrigerated or is located in air circulating passages communicating with such spaces.

displacement—(1) volume swept by piston per stroke. (2) Actual—the actual volume of gas at compressor inlet conditions moved in a given time, or (3) theoretical – the total volume displaced by all the pistons of a compressor per stroke in a definite time interval. Usually measured in cubic feet per minute.

distributor—device for dividing flow of liquid fluid between parallel paths in an evaporator, or in other types of heat transfer apparatus.

domestic refrigerator—refrigerator or mechanical refrigerator for a home, usually of size of 16 cu ft or less.

double-pipe heat exchanger—one in which two pipes are arranged concentrically, one within the other, and in which one fluid flows through the inner pipe and the other through the annulus between them.

double suction—split suction, valving arrangement on compressors for carrying two suction pressures, see multiple effect.

drier—a chemical compound capable of adsorbing, or reacting chemically with the moisture contained in the liquid or gaseous refrigerant-oil mixture.

drip—water which appears on thawing frozen food, water melting from evaporator, water dropping from a cooling surface.

dry air—air without its contained water vapor; air only.

dry-air cooler—removes sensible heat from the dehydrated air whenever it leaves the dehydrator at an elevated temperature.

dry bulb—temperature, temperature by ordinary thermometer (term used only to distinguish from wet bulb temperature).

dry ice—solid carbon dioxide, CO_2 (proprietary term).

dry tons—the sensible heat load expressed in tons of refrigeration.

dual compression—see multiple effect.

dual effect control—one responsive to temperatures of two zones, or to two variable conditions.

dual-temperature brine system—in chilling beef the use of an initial brine temperature, followed by the lower brine temperature.

duct—a conduit or tube used for conveying air or other gas.

duct system—a series of ducts, elbows and connectors to convey air from one location to another.

dust—solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

dynamometer—device for applying and measuring power developed by an engine or motor.

ebullator—a device inserted in flooded evaporator tubes to prevent the evaporator from becoming oil bound or refrigerant liquid becoming quiescent at a pressure lower than its boiling point.

effective temperature difference—the difference between the room air temperature and the supply air temperature at the outlet to the room.

effective work—net mechanical energy required by, or load imparted to, piston of compressor.

efficiency, compression—ratio of hp required to compress adiabatically and reversibly all the vapor delivered by compressor (in single stage) to the ihp.

efficiency, mechanical—ratio of the compression energy or work of a compressor to the energy or work input.

efficiency, volumetric—(1) *Total*—ratio of the actual volume of gas moved by the compressor or pump to the actual displacement of the compressor or pump. (2) *Apparent*-ratio of length on indicator card of suction line to stroke. (3) *Due to cylinder heating* (also called real or no-clearance volumetric efficiency)-ratio of the total to the apparent volumetric efficiency.

ejector—a device which builds up a high fluid velocity in a restricted area to obtain a lower static pressure at that point so that fluid from another source may be drawn in.

electric precipitators—devices for removing dust from the air by means of electric charges induced on the dust particles.

electric refrigerator—refrigerator, cooled by a system, operated by electric motor as distinct from an engine-operated, or a gas-operated or brine-cooled refrigerator.

electrical degree—the 360th part of the angle subtended at the axis of the machine, by two consecutive field poles of like polarity. One mechanical degree is thus equal to as many electrical degrees as there are pairs of poles in the machine.

emissivity—characteristic of a surface for giving off heat by radiation. Almost equal to the absorptivity. Ranges in value from zero for a perfect reflector to unity for a perfect black body.

emulsification—formation of an emulsion, i.e., a mixture of small droplets of two or more liquids which do not dissolve in each other.

enclosed vertical compressor—machine with vertical cylinder and enclosed crank, generally single-acting.

engine—prime mover, device for transforming fuel or heat energy into mechanical energy.

enthalpy—a thermodynamic property of a substance defined $h = u + Pv/J$ where u = internal energy, customarily in Btu per lb; P = pressure in lb per sq ft; and v is specific volume in cu ft per lb. $J = 778$ ft lb per Btu. Also called *total heat* or *heat content*.

entropy—entropy is the ratio of the heat added to a substance to the absolute temperature at which it is added for a reversible process. It is a thermodynamic property which, for practical purposes, is best defined by stating its principal functions: (1) during a reversible adiabatic change of state, entropy is constant; (2) during a reversible isothermal change of state, the heat absorbed is equal to absolute temperature times change of entropy.

enzyme—complex organic substances such as diastase, pepsin, etc., capable of transforming by catalytic action some other compound; a soluble ferment.

equalizer—piping arrangement on vertical enclosed compressors to equalize refrigerant gas pressure in crank case and suction; device for dividing the liquid refrigerant between parallel low side coils.

equivalent evaporation—the amount of water a boiler would evaporate, in pounds per hour, if it received and vaporized feed water at 212 F and atmospheric pressure as compared to actual evaporation.

eutectic mixture or solution—a mixture which melts or freezes completely at constant temperature and with constant composition. Its melting point is the lowest possible for mixtures of the given substances.

evaporation—change of state from liquid to vapor.

evaporative condenser—condenser in which heat absorbed by water is largely dissipated as latent heat by evaporation of the water into an unsaturated air stream in contact with it.

evaporative cooling—involves the adiabatic exchange of heat between air and a water spray or wetted surface. The water assumes the wet bulb temperature of the air, which remains constant during its traverse of the exchanger.

evaporative equilibrium—of a wet bulb instrument—the condition attained when the wetted wick has reached a stable and constant temperature (when the instrument is exposed to air at velocities over 900 fpm this temperature may be considered to approach the true wet bulb temperature).

evaporator—that part of a system in which refrigerant is vaporized to produce refrigeration.

evaporator, dry-type—an evaporator of the continuous tube type where refrigerant from a pressure-reducing device is fed into one end and the suction line connects to the outlet end.

exfiltration—air flow outward through a wall, leak, membrane, etc.

exhaust opening—any opening through which air is removed from a space which is being heated or cooled, or humidified or dehumidified, or ventilated.

expansion coil—an evaporator constructed of pipe or tubing.

extended surface—heat transfer surface one side of which is increased in area by the addition of fins, discs or other means.

external equalizer—in a thermostatic expansion valve, a tube connection from the chamber containing the evaporation pressure actuated element of the valve to the outlet or the evaporator coil. A device to compensate for excessive pressure drop through the coil.

extruded—pushed out through a die. Bars of ice, metal rods, shapes and tubes are made by this method.

f—film or surface conductance, the amount of heat expressed in Btu transmitted by radiation, conduction, convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance. The letter f is also used for the friction factor in fluid flow.

factor of safety—ratio of ultimate stress to designed working stress.

fahrenheit—a thermometric system in which 32 deg denotes freezing and 212 deg the boiling point of water under normal pressure at sea level (14.696 psi).

fan—strictly, device for imparting motion to air. Any device for imparting motion to air by means other than positive displacement.

fan, attic—fan, mounted in roof or side wall for exhausting air from attic area.

fan economizers—devices which prevent the operation of the fan motor on a cold diffuser during the shut-down period after the coil has been defrosted.

fan exhaust—fan used to remove or exhaust air from a space.

filter—a device to remove solid material from a fluid.

filter press—a device for separating solid and liquid matter under pressure so that the solid residue is compressed into briquettes to facilitate removal.

fin—an extended surface to increase the heat transfer area, as metal sheets attached to tubes.

finned tubes—heat transfer tube or pipe with extended surface in the form of fins, discs or ribs.

first and second laws—of thermodynamics—first law states, in effect, the conservation of energy principle, particularly equating heat and mechanical energy and denying perpetual motion, insofar as it implies a creation of energy. Second law states that the quality of energy varies as to form, particularly that heat energy is only in part transformable into mechanical energy; it denies the possibility of a machine operating in a cycle and developing mechanical energy from a single source of heat.

flammability—the ability of a material to burn.

flammable refrigerant—any refrigerant which will burn when mixed with air, such as ethyl chloride, methyl chloride and the hydrocarbons.

flare fitting—a type of soft-tube connector which involves the flaring of the tube to provide a mechanical seal.

flash gas—the gas resulting from the instantaneous evaporation of refrigerant in a pressure-reducing device to cool the refrigerant to the evaporation temperature obtaining at the reduced pressure.

flash chamber—separating tank placed between the expansion valve and evaporator in a refrigeration system to separate and by-pass any flash gas formed in the expansion valve.

flash point—temperature of combustible material, as oil, at which there is a sufficient vaporization to ignite the vapor, but not sufficient vaporization to support combustion of the material.

flooded system—system in which only part of the refrigerant passing over the heat transfer surface is evaporated, and the portion not evaporated is separated from the vapor and recirculated.

flotation—a method of treating materials by floating in a liquid. An ice-making method in which ice floats away from the surface on which it has been frozen.

fluid—gas, vapor or liquid.

foaming—formation of a foam or froth of oil-refrigerant due to rapid boiling out of the refrigerant dissolved in the oil when the pressure is suddenly reduced. This occurs when the compressor starts operating and, if large quantities of refrigerant have been dissolved, large quantities of oil may "boil" out and be carried through the refrigerant lines.

forecooler—in an ice plant, device for cooling the water for ice-making before its enters the cans, precooler.

foul gas—see non-condensible gas.

free area—the total minimum area of the openings in a grille, face, or register through which air can pass.

free delivery type unit—a device which takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

freezer—in cold storage, an insulated room kept below 30 F—see (a) carrying freezer, (b) sharp freezer. Any device for freezing perishables.

freeze up—failure of a refrigeration unit to operate normally due to formation of ice at the expansion valve. The valve may be frozen shut or open, causing improper refrigeration in either case.

freezing point—temperature at which given liquid substance will solidify or freeze upon removal of heat. Ice point for water is 32 F.

freezing tank—a container kept at low temperatures for storing foodstuffs or for freezing foodstuffs.

freezing time—time for any complete freezing process to take place.

Freon-12—the trade name for dichlorodifluoromethane (CCl_2F_2), also called F-12.

frost back—the flooding of liquid from an evaporator into the suction line accompanied by frost formation on suction line in most cases.

frozen food—any food in which the contained water is in the solid form.

fusible plug—a device having a predetermined melting temperature member for the relief of pressure.

gage (gauge)—instrument for measuring pressure or liquid level. Also arbitrary scale of measurement for sheet metal thickness, wire and drill diameters, etc.

gage glass—device for showing a liquid level.

gas constant—the coefficient R in the perfect gas equation $pv = RT$.

gas refrigerator—refrigerator motivated by thermal energy of burning gas.

gaseous state of matter—one of three states characterized by greatest freedom of molecules and lack of any inherent fixed shape or volume.

generator—basic part of absorption system; a still provided with means of heating used to drive ammonia out of water.

glazing—of foods, freezing a coat of ice on frozen food by dipping in water, the latent heat of the coat being absorbed inwards.

graduated acting—term applied to a control instrument or device which functions to give throttling control, that is, operates between full on and full off position.

grid—a device for lifting a row of ice cans.

grille—a lattice or grating for delivery or intake opening of an air passage.

head—in hydraulics, the height of a column of fluid necessary to produce existing pressure at a point in the system. Also velocity head the pressure equivalent to fluid velocity.

head pressure—*operating pressure* measured in the discharge line at the outlet from the *compressor*.

heat of condensation—latent heat given up in changing from gas to liquid.

heat exchanger—apparatus in which heat is exchanged from one fluid to another through a partition.

heat, latent—heat characterized by a change of state of the substance concerned, for a given pressure and always at a constant temperature for a pure substance, i.e., heat of vaporization or of fusion.

heat of the liquid—enthalpy of a mass of liquid above an arbitrary zero.

heat of reaction—heat per unit mass or per mole of one of the reagents or products of reaction in a chemical reaction; exothermal if given off, endothermal if absorbed.

heat pump—a refrigerating system employed to transfer heat into a space or substance. The condenser provides the heat while the evaporator is arranged to pick up heat from air, water, etc. By shifting the flow of air or other fluid a heat pump system may also be used to cool the space.

heat rejection effect, condensing—that portion of the total refrigerant heat rejecting effect of a condenser which is used for condensing the entering refrigerant vapor to a saturated liquid at the entering refrigerant pressure.

heat rejection effect, total refrigerant—total useful capacity of a refrigerant condenser for removing heat from the refrigerant circulated through it.

heat transmission—any flow of heat; usually refers to conduction, convection and radiation combined.

hermetically-sealed unit—a sealed (hermetic-type) condensing unit is a mechanical condensing unit in which the compressor and compressor motor are enclosed in the same housing, with no external shaft or shaft seal, the compressor motor operating in the refrigerant atmosphere. The compressor and compressor motor housing may be of either the fully-welded or brazed type, or of the service-sealed type. In the fully-welded or brazed type, the housing is permanently sealed and is not provided with means of access for servicing internal parts in the field. In the service-sealed type, the housing is provided with some means of access for servicing internal parts in the field.

high side—parts of a refrigerating system under condenser pressure or higher.

hold-over—in an evaporator, the ability to stay cold after heat removal from the evaporator stops. A material used to store heat in latent or sensible form.

horizontal compressor—machine with horizontal cylinder, generally double-acting.

horsepower—unit of power in foot-pound-second system, work done at the rate of 550 ft lb per sec, or 33,000 ft lb per min.

hot-gas defrosting—use of high pressure or condenser gas in the evaporator or low side to effect removal of frost.

humidifying effect of a unit in Btu per hr—the latent heat of vaporization of water at the average evaporating temperature times the number of pounds of water evaporated per hour.

humid heat—increase of enthalpy per pound of dry air with its associated moisture under conditions of constant pressure and constant specific humidity.

humidifier—a device to add moisture to air.

humidifier—central—a device which humidifies air to be circulated through ducts in an air-conditioning system.

humidifier—room-spray type—an air humidifier which sprays water directly into the room.

humidify—to add water vapor to the atmosphere; to add water vapor or moisture to any material.

humidistat—control device responsive to atmospheric relative humidity, hygrostat.

humidity, absolute—concentration of water vapor in unit volume, as grains of H_2O per cu ft or per cu ft of air.

humidity ratio—see specific humidity.

humidity, relative—the ratio of the partial pressure of water vapor in the air to the pressure of saturated water vapor at the temperature of the air. This is closely equivalent at temperatures below 150 F to a more rigorous definition—the ratio of the density of water vapor in the air to the density of saturated water vapor at the air temperature.

humidity, specific—the weight of water vapor (steam) associated with one pound weight of dry air, sometimes called *humidity ratio.*

humidity, percentage—the ratio of the weight of water vapor mixed with a pound of dry air to the weight of saturated water vapor at the same temperature mixed with a pound of dry air, expressed in percent. This ratio of specific humidities is also called *degree of saturation* or *saturation ratio.*

hydrator—a covered container or compartment in a refrigerator having a higher relative humidity than the balance of the storage space.

hydrolysis—reaction of a material such as Freon-12 or methyl chloride with water. Acid materials in general are formed.

hydrometer—an instrument which, by the extent of its submergence, indicates the specific gravity of the liquid in which it floats.

hydrostatic pressure—the pressure due to the vertical height of a liquid column.

hygrometer—instrument responsive to humidity conditions (usually relative humidity) of the atmosphere.

hygroscopic—absorptive of moisture, readily absorbing and retaining moisture.

hygrostat—automatic control responsive to humidity.

ice box—refrigerator using ice as the primary cooling medium.

ice can—an open content top tank or can of properly formed and reinforced sheet metal which when submerged in a chilled anti-freeze solution in an ice tank will permit the freezing of its water into a block of ice of a desired form or weight.

ice-can frames—a metal frame or support in which a number of ice cans are fixed so as to enable them to be handled as a group rather than handling each can individually.

ice-can truck—a wheeled frame with arm and hoist suitably arranged to be moved over the ice-tank top for lifting individual cans from the tank.

ice crane, hoist and runway—a device consisting of a crane beam, with suitable carriage, carrying a movable hoist for lifting the ice cans or ice blocks, individually or in groups, and running on suitably supported runway rails all for the purpose of lifting the cans or ice blocks from the tank and carrying them to a point beyond the tank for disposal.

ice-making capacity—actual productive ability of a system making ice. This is less than the rated (ice-melting) capacity as some refrigeration is used in cooling the water to freezing point, cooling the ice below the freezing point, and overcoming heat leakage.

ice-making tank—a tank arranged with proper accessories to hold an evaporator and anti-freeze solution and ice cans, the cans being immersed in the solution which usually is in circulation around the evaporator and the cans.

ice-melting capacity—1. Refrigeration equal to the latent heat of fusion of a stated weight of ice at 144 Btu per lb. 2. Ice-melting equivalent—the amount of heat absorbed by one pound of ice at 32 F in liquefying to water at 32 F, 144 Btu.

ice point—temperature at which water freezes under normal atmospheric pressure (14.696 psia), +32 F, 0 C.

ice-tank grating and covers—a framework of wood, metal or combination of each, which positions the ice cans in the ice-making tank and upon which are supported removable covers, usually of wood, above the cans excluding dirt and the warmer air of the tank room.

indicated work—work equivalent of indicator card area for a reciprocating compressor or engine.

indicator diagram—pressure-volume diagram tracing condition of gas in a compressor or engine cycle, in terms of p and v; indicator card.

indirect method of refrigeration (or cooling)—a system in which a liquid, such as brine or water, cooled by the refrigerant, is circulated to the material or space refrigerated or is used to cool air so circulated.

induction units—high-pressure type—employ nozzles which produce a high velocity jet. The high velocity jet of primary air induces a flow of secondary air through coils located in the secondary air stream.

induction units—low-pressure type—essentially induction type convectors. They use a jet of conditioned air (or primary air) to induce into the unit a flow of room or secondary air which mixes with the primary air. The mixture is discharged into the room through a grille at the top of the unit. Heating coils are located in the secondary air stream for use in heating.

industrial air conditioning—air conditioning for other uses than comfort.

industrial system—a system used in the manufacture or processing of materials, such as ice-making plants, cold storage warehouses, ice cream plants, dairy plants, packing houses, chemical plants, and other places of similar industrial enterprise.

inert gas—one that experiences no change of state in a system or process, such as air, nitrogen, helium, etc., mixed with a volatile refrigerant.

infiltration—air flowing *inward* as through a wall, leak, etc.

inlet—suction, low side-pipe to, or port of compressor.

insulation, fill—granulated, shredded, or powdered materials, prepared from vegetable, animal, or mineral origin. It can come in bulk or batt form.

insulation, sound—acoustical treatment of fan housings, supply ducts, and other parts of system and equipment for isolating of vibration.

intercooling—removal of heat from compressed gas between compression stages.

internal (intrinsic) energy—the sum of all the kinetic and potential energies contained in a substance due to the states of motion and separation of its several molecules, atoms and electrons. It includes sensible heat (vibration energy) and that part of the latent heat that is represented by the increase in energy during evaporation.

irradiation—subjecting foods, etc., to radiations of special wavelengths, such as the 0.2537 micron wavelength, which kills certain bacteria.

is-, iso- —prefix meaning constant; as isothermal, constant temperature; isentropic, constant entropy; isobaric, constant pressure; etc., and in chemicals one having different characteristics but with the same number and kind of atoms.

jacket water—in a compressor, the water used for cooling the cylinder head and/or walls.

jacketing—surrounding by a confined bath or stream of fluid for temperature control or heat absorption.

Joule-Thomson effect—dT/dp of an actual gas in a process of throttling or expansion without doing work or interchanging heat.

"k" factor—thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two sides of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

lag of temperature control—the delay in action of a temperature responsive element due to time required for the element to reach surrounding temperature.

latent heat—change of enthalpy during a change of state usually expressed in Btu per lb. With pure substances, latent heat is absorbed or rejected at constant temperature at any pressure.

leak detector—1. A device used to detect refrigerant leaks in a refrigerating system, or 2. Halide torch—a flame tester generally using alcohol and burning with a blue flame. When the sampling tube draws in Freon vapor, color of flame changes to bright green.

line—mainly used in reference to a line on characteristic psychrometric chart or indicator—liquid, pipe conveying liquid, etc.

liquefaction—change of state to liquid, generally used instead of condensation in case of substances ordinarily gaseous.

liquid line—the tube or pipe carrying the refrigerant liquid from the condenser or receiver of a refrigerating system to a pressure-reducing device.

liquid receiver—a vessel permanently connected to a system by inlet and outlet pipes for the storage of (liquefied) refrigerant.

liquid state of matter—one of three states characterized by limited freedom of molecules and by substantial incompressibility.

liquor—a solution in refrigeration, generally of ammonia and water.

load—the amount of heat per unit time imposed on a refrigerating system, or the required rate of heat removal.

load factor—ratio of actual mean load to a maximum load or maximum production capacity in a given period.

locker plants—cold storage establishments containing food storage boxes or lockers for individual users.

louvres—sloping overlapping boards or metal plates intended to permit ventilation and shed falling water.

low-pressure control—a thermostat (usually switch) actuated directly by refrigerant vapor pressure in the low side.

low-pressure system—in air conditioning, a distributing system delivering air to ordinary ventilating grilles at low velocities with low static losses through the supply grilles.

low side—parts of a refrigerating system under evaporator pressure.

low temperature—temperature below ordinary refrigerating plant requirements—say, from −30 F on down; also any part of a system below another parallel refrigerating level of temperature.

lyophilization—the process of dehydrating a frozen substance under conditions of sublimation.

machinery room—a room in which is permanently installed and operated a refrigerating system but not including evaporators located in a cold storage room refrigerator box, air-cooled space, or other enclosed space. Closets solely contained within and opening only into a room shall be considered a part of such room and not as a machinery room.

main—pipe for distributing to or collecting from various branches.

makeup water—water supplied to replenish as water replacing that lost by evaporation.

manifold—portion of main in which several branches are close together. Also, single piece in which there are several fluid paths.

manometer—U-tube for measuring pressure differences.

mean temperature difference—mean of difference between temperatures of a fluid receiving and a fluid yielding heat.

mechanical condensing unit—a complete high side of a refrigeration system including the motor in a unit assembly.

mechanical equivalent of heat—an energy conversion ratio of 778.16 foot pounds = 1 Btu.

mechanical draft cooling tower—cooling tower through which air is forced or induced by fans or other mechanical means.

melting—change of state from solid to liquid.

melting point—for a given pressure, the temperature above which solid state cannot exist.

metabolism—chemical changes in living cells by which energy is provided for vital processes.

meter—unit of length = 39.37 inches. Also instrument for measuring rates or integrating rates over a period of time.

micron—a unit of length, the thousandth part of one millimeter or the millionth of a meter.

modulating—of a control, tending to adjust by increments and decrements, also one modified by variation of a second condition.

moisture tons—latent heat load expressed in tons of refrigeration.

mole (mol)—weight of a substance equal, in the weight units used, to its molecular weight; as a gram-mole; pound mole, etc.

motive power—term used to express the source from which a device such as a diaphragm valve, relay or motor obtains its power (electric or fluid).

motor, air—an air-operated device which is used primarily for opening or closing dampers.

multiple can filler—an elevated tank having a compartment for each ice can to be filled. Each compartment is provided with adjusting devices so that it will be filled with a predetermined weight of water. The water is transferred to the ice cans through valved pipes.

multiple system—a system using the direct method in which refrigerant is delivered to two or more evaporators in separate rooms or refrigerators.

multi-stage compression—in 2 or more steps, as where the discharge of one compressor is connected with the suction of another.

multi-stage expansion—passing volatile refrigerant through 2 or more pressure-reducing devices, connected in series, usually with an evaporator between them operating at one pressure and a second evaporator fed through both devices, at a lower pressure.

natural-draft cooling tower—a cooling tower in which air is circulated by natural convection.

non-condensible gas—gas in a refrigerating system which does not condense at the temperature and partial pressure at which it exists in the condenser, and therefore imposes a higher head pressure on the system.

non-priming cooler—tubes are omitted from the top segment of the shell leaving a gas space above the tubes about equal to ½ the inside shell diameter.

off-peak system—one with control which normally avoids use of power during peak load periods, usually having eutectic or water-ice hold-over means.

oil cooler—heat exchanger for the purpose of cooling oil in a lubrication system.

oil separator—device for preventing oil from entering the evaporator.

open shell-and-tube condenser—one in which the water passes in a film over the inner surfaces of the tubes, which are open to the atmosphere.

open-type compressor (or system)—A compressor with a shaft or other moving part extending through a casing to be driven by an outside source of power, thus requiring a stuffing box, shaft seal or equivalent rubbing contact between a fixed and moving part.

organic compound—originally a chemical compound produced by a life process. Now generally understood to include all compounds containing carbon.

outlets, ceiling—outlets installed in ceiling in order that a blanket of cool air be introduced at the top of the room, gradually settling evenly over the occupied zone.

outlets slotted—flat metal plates containing a number of long narrow slots or a single long narrow slot.

outlets, vaned—equipped with vertical and/or horizontal adjustable vanes.

output—capacity, duty, performance, net refrigeration produced by system, equal to load imposed. Motor output is electrical power input times motor efficiency.

outside air—external air, atmosphere exterior to refrigerated or conditioned space, ambient (surrounding air).

outside air opening—any opening used as an entry for air from outdoors.

over-run—in ice cream freezing, the ratio of the volume of ice cream to the volume of the mix used.

ozone—triatomic oxygen, O_3, sometimes used in air conditioning or cold storage as an odor eliminator, somewhat toxic.

packing—the stuffing around a shaft or valve stem to prevent fluid leakage.

packing plant—an establishment engaged in the slaughtering, dressing and processing of animals. Also used in connection with the processing of vegetables or fish.

parallel, connection in—system whereby flow is divided among 2 or more channels, from a common starting point or header.

pasteurization—heat treatment, usually at 131–158 F, for killing bacteria as in milk without greatly changing its chemical composition.

performance factor—the ratio of heat moved by a refrigerating system to the heat equivalent of the energy used.

phase—in thermodynamics one of the states of matter, as solid, liquid, gaseous; electrical, an alternating current whose alternations have a definite time relation to the rotational position of the alternator. In a polyphase machine, the phases are separated by 360 electrical degrees divided by the number of phases.

phosphorous pentoxide—an efficient drier material which becomes gummy on reacting with moisture and hence is not used alone as a drying agent in refrigerating systems.

pipe line refrigeration—service to group of buildings with a refrigerant supply from a central refrigerating plant.

plenum—chamber under pressure for receiving air before delivery to conditioned space or combustion system.

pneumatic—operated by air pressure.

polytropic-change—any set of changes in a gas represented by the equation $pV^n = \text{constant}$.

positive displacement compressor—a compressor in which increase in vapor pressure is attained by changing the internal volume of the compression chamber.

potentiometer—an instrument for measuring or comparing small electromotive forces such as those produced by thermocouples.

power—the rate of doing work usually expressed in horsepower, kilowatts, etc.

power consumption—power used multiplied by time, measured in kw hr, hp hr, etc. (Power is the rate of work; power consumption is work.)

precooler—cooler for the removal of sensible heat before shipping, storing, or processing; device for cooling a fluid before it enters some piece of apparatus.

precooling—refers to practice of preparing fruits and vegetables for shipment by cooling before or after they enter car.

preheating—in air conditioning to heat the air in advance of other processes.

pressure—(1) thermodynamically the normal force exerted by a homogeneous liquid or gas, per unit of area, on wall of the container. (2) Absolute—pressure referred to that of a perfect vacuum. It is the sum of gage pressure and atmospheric pressure. (3) Atmospheric—pressure exerted by atmosphere in all directions as indicated by a barometer. Standard atmospheric pressure is a pressure of 76 cm of mercury at 32 F. It is equivalent to 14.6959 psi.

(4) Back—or suction—pressure in suction line of evaporator. (5) Balance—pressure in a system or container equal to that which exists outside. (6) Critical—vapor pressure corresponding to the critical temperature. (7) Gage—pressure above atmospheric. (8) Partial—portion of total gas pressure of a mixture attributable to one component.

pressure drop—loss in pressure, as from one end of a refrigerant line to the other, due to friction, etc.

pressure equalizing—allowing high and low-side pressures to equalize or nearly equalize during idle periods as by use of an unloading valve or a vapor lock liquid control; or nearly equalizing inlet and discharge pressures on the compressor. In either case, to reduce starting torque load.

pressure-imposing element—every device or portion of the equipment used for the purpose of increasing the pressure upon the refrigerant.

pressure-limiting device—a pressure-responsive mechanism designed to stop automatically the operation of the pressure-imposing element at a predetermined pressure.

pressure relief device—a valve or rupture member designed to relieve excessive pressure automatically.

pressure relief valve—a valve held closed by a spring or other means and designed to relieve automatically pressure in excess of its setting.

pressure vessel—any fluid-containing receptacle. In a refrigerating system other than expansion coils, headers and pipe connections.

primary fluid—the refrigerant, to distinguish from secondary fluid, or brine.

psychrometer—instrument for measuring relative humidities by means of wet and dry bulb temperatures. See hygrometer.

psychrometric chart—a graphical representation of the properties of steam and air mixtures.

psychrometry—use of psychrometer, but broadly speaking the science and practices associated with atmospheric air mixtures and their control.

pump down (refrigeration system)—the operation by which the refrigerant in a charged system is pumped into the liquid receiver.

purging—the act of blowing out gas from a refrigerant-containing vessel, usually for the purpose of removing non-condensibles.

purger—device for removing non-condensible gas from refrigerant condensers or for removing low concentration liquor from absorption system evaporators.

pyrometer—instrument for measuring high temperature.

quality, of wet vapor—fraction by weight of vapor in mixture of liquid and vapor.

quick freezer room—room kept at very low temperature for the purpose of freezing foodstuffs rapidly.

radiation—the transmission of heat through space by wave motion. The passage of heat from one object to another without warming the space between. The heat is passed by wave motion similarly to transmission of light.

ratio of compression—ratio of absolute pressures after and before compression.

raw water—in ice making, any water used for ice making except distilled water.

reach-in refrigerator—commercial refrigerator, service refrigerator.

reciprocating compressor—a positive displacement compressor with a piston or pistons moving in a straight line but alternately in opposite directions.

recirculated air—return air passed through the conditioner before being again supplied to the conditioned space.

rectifier—in refrigeration, externally cooled heat exchanger in high side of absorption system for condensing absorbent and separating it from refrigerant before passing to condenser. In electricity, a device for converting AC to DC current.

reexpansion line—curve on indicator card, representing path of fluid left in cylinder at end of stroke during return of piston prior to inlet.

refrigerant—the medium of heat transfer in a refrigerating system which picks up heat by evaporating at a low temperature and pressure and gives up heat on condensing at a higher temperature and pressure.

refrigerated plate method of freezing—heat transfer through the direct contact of refrigerated plates with the packaged product.

refrigerating engineering—technique of design, manufacture, application and operation of refrigerating machinery and its primary equipment. Refrigeration (except as exact measure in heat units) refers here to a more general science, more concerned with the use of coldness for commercial and other useful purposes.

refrigerating machine—broadly, the refrigerant system proper, as condenser, receiver, compressor, and evaporator of compression system; strictly the high side only. Also used to refer to compressor only.

refrigerating medium—this may mean refrigerating fluid, but usually refers to a cold medium such as ice, solid CO_2, frozen brine, etc.

refrigerating system—a combination of interconnected refrigerant-containing parts in which a refrigerant is circulated for the purpose of extracting heat.

refrigerator—a container and means of cooling it (domestic) or a large container such as a storage refrigerator, service refrigerator, etc.

regain—of moisture, the amount absorbed by any material in percent of weight of that material.

regelation—refreezing of water that has resulted from the melting of ice under pressure; does not require refrigeration.

regenerative heating (or cooling)—process of utilizing heat which must be rejected in one part of the cycle, to perform a useful function in another part of the cycle, by heat transfer.

register—grille supplied with control damper.

reheating air—in an air conditioning system the final step in treatment, in the event the temperature is too low.

resistance, thermal—the reciprocal of conductance, or of overall coefficient of heat transfer.

respiration—production of CO_2 and heat by ripening of perishables in storage, also the breathing process of animals.

return air—air returned from conditioned or refrigerated space.

reverse cycle—heating—see heat pump.

reversible cycle—theoretical thermodynamic cycle which can be completely reversed; e.g., Carnot cycle.

room dry bulb (dew point, etc.)—the dry bulb (dew point, etc.) temperature of the conditioned room or space.

room air conditioner—air conditioning element for a room, self-contained system for conditioning atmosphere for comfort.

rotary compressor—one in which compression is attained in a cylinder by rotation of a positive displacement member.

run-around system—may be direct air-to-air heat exchangers or conduits in which is circulated brine, water, or expanding and condensing refrigerant. Apparatus consists of a surface or spray dehumidifier with counterflow water (or brine) coils on both entering and leaving sides, connected in series with a circulating pump. Heat from the entering air is absorbed by the precooler and returned to the leaving air by the after-heater coil.

running out—of food juices, see drip.

rupture member—a device which will automatically rupture at a predetermined pressure.

safety head—in a compressor, a cylinder head held in place by a spring of such strength that it will not be compressed during normal operation, but will be moved by solid or liquid matter or abnormal gas pressure between it and the piston, thereby protecting the compressor.

salinometer—hydrometer calibrated in salt concentration.

saponify—turn to soap, as of oil in contact with refrigerant; properly, the decomposition of ester into an alcohol and an acid.

saturated air—air in which the partial pressure of water vapor is equal to the vapor pressure of water at the existing temperature.

saturated vapor—vapor in equilibrium with its liquid, that is, when the number of molecules passing in two directions through the surface dividing the two phases is equal.

saturation—the condition for coexistence in stable equilibrium of two or more distinct phases.

saturation, degree of, μ—the ratio of the specific humidity of an air water-vapor mixture to the specific humidity of this mixture saturated with water vapor at the same temperature.

seal, bellows—metal bellows used in a shaft seal or in place of a packing for valves. Also used in long pipe lines instead of gaskets to compensate for expansion of the line with temperature.

sealed unit—a motor compressor assembly having the compressor contained within a gas-tight casing through which no shaft extends. Drive is usually by a motor within the same casing, but may be induced by external means.

secondary condenser—the condenser of a secondary system; also, a condenser and a secondary system where the condenser is cooled by the evaporator of the secondary system.

secondary fluid—fluid cooled by the refrigerant in indirect method of refrigeration.

secondary refrigerant—the volatile refrigerant used in a secondary system or a non-volatile liquid refrigerant such as brine. It might be air. Should be further defined as volatile, non-volatile or gaseous.

self-contained air conditioning or cooling unit—a condensing unit combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified according to the method of rejecting condenser heat (water-cooled, air-cooled and evaporatively cooled), method of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).

self-contained system—a complete factory-made and factory-tested system in a suitable frame or enclosure which is fabricated and shipped in one or more sections and in which no refrigerant-containing parts are connected in the field other than by companion or block valves. See also unit system.

sensible heat—heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.

sensible heat coolers—in the form of cooling surface using water, brine or direct expansion refrigerant. It is always located on the leaving side or the dehydrator, but frequently treats in addition a large volume of room air which is not circulated through the dehydrator for moisture reduction.

service refrigerator—commercial refrigerator of reach-in type.

shaft seal—a rubbing seal or stuffing box—used to prevent fluid leakage between the shaft and bearing of a compressor or other fluid moving device.

sharp freezer—cold storage freezer room, generally kept at -30 F to -10 F, to receive unfrozen goods and freeze them.

shell-and-tube—pertaining to heat exchangers in which a nest of tubes or pipes, or a coil of tube or pipe, is contained in a shell or container. The pipe (or pipes) carries a fluid through it while the shell is also provided with an inlet and outlet for a fluid flow.

shell-type apparatus—a refrigerant-containing pressure vessel having tubes for the passage of a refrigerating fluid or one to be cooled.

sight glass—glass tube used to indicate the liquid level in tanks, bearings and similar equipment.

silica gel—a drier material having the formula, SiO_2.

single can fillers—a device connected by hose with the source of water supply and provided with a starting device and float actuated valve, such that when placed in an ice can and started, it will deliver to the can a predetermined volume of water and then stop the flow.

sludge—a decomposition product formed in a system due to impurities in the oil or to moisture. Sludge may be gummy or hard.

small machine—compressor, high side or complete system having small capacity; commercial machine.

solenoid valve—a valve which is closed by gravity, pressure, or spring action and opened by the movement of a plunger due to the magnetic action of an electrically energized coil. A valve which is normally open and becomes closed when energized will be known as a *reverse-acting solenoid valve.*

solubility—of solid, liquid or gas A in solid or liquid B, the mass of A, per unit mass of B, contained in a saturated solution of A in B.

specific gravity—density compared to density of standard material; reference usually to water or to air.

specific heat—energy per unit of mass required to produce one degree rise in temperature, usually Btu per lb deg F numerically equal to cal per gram deg C.

specific volume—volume per unit of mass, usually cu ft per lb.

spray deck—overhead bunker where air is cooled and circulated by brine sprays.

spray method of freezing—refrigerated liquid is sprayed into an insulated enclosure containing the product to be frozen.

spray pond—arrangement for lowering the temperature of water by evaporative cooling, in contact with outside air, in which the water to be cooled is sprayed by nozzles into the space above a body of previously cooled water, and allowed to fall by gravity into it.

standard air—air weighing 0.075 lb per cu ft which is closely air at 68 F dry bulb and 10% relative humidity at a barometric pressure of 29.92 in. of mercury, or approximately dry air at 70 F at the same pressure.

standard conditions—operating conditions specified by codes under which standard ratings are obtained.

standard rating—of a refrigerating machine, is capacity, in tons or Btu per hr, when operating under standard conditions.

steam-jet system—high-pressure steam supplied through a nozzle and acting to eject vapor from the evaporator, thus maintaining the requisite low pressure on one side and producing a higher pressure on the other by virtue of compression in a following diffusion passage.

stop valve—a shut-off for controlling the flow of refrigerant.

storage room—any cold storage space, usually on larger plant scale.

stroke—of piston, length of travel, twice its crank radius.

strong, rich—liquor or solution, water with high concentration of another substance, as ammonia, salt, etc.

stuffing box—a packing gland surrounding a shaft, stem or rod to prevent leakage.

subcooling—process of cooling refrigerant below condensing temperature, for a given pressure. Also cooling a liquid below its freezing point, where it can exist only in a state of unstable equilibrium.

subcooling refrigeration effect—additional refrigeration effect made available by subcooling the refrigerant liquid.

sublimation—change of state directly from solid to gas without appearance of liquid.

submerged condenser—condenser piping submerged in bath of condenser water.

suction line—the tube or pipe which carries the refrigerant vapor from the evaporator to the compressor inlet.

summer air conditioning—cooling, dehumidifying, air distribution and air cleaning, when outside temperatures are above the inside or room temperature.

sun effect—solar energy transmitted into space through windows and building materials.

superheated vapor or gas—vapor at a temperature which is higher than the saturation temperature (i.e., boiling point) at the existing pressure.

surface conductance—the amount of heat transmitted by radiation, conduction, and convection from a surface to the air or liquid surrounding it, or vice versa, in one hour per square foot of surface for a difference in temperature of 1 deg F between the surface and the surrounding air or liquid.

surface cooling—a method of cooling air or other gas by passing over cold surfaces.

surge drum—see accumulator.

sweating—condensation of moisture from air on a surface which is below the dew point temperature.

system—a refrigerating scheme and/or machine usually confined to those parts in contact with refrigerant; an arbitrarily chosen group of materials and devices set apart for analytical study.

temperature, effective—an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air.

temperature, room—temperature of any room, as for example (1) a room in which refrigerator is being operated or tested; (2) a room being conditioned for the comfort of occupants. Colloq: used to mean ordinary temperature one is accustomed to find in dwellings.

temperature, saturation—of a fluid, boiling point corresponding to a given pressure; evaporation temperature; condensation temperature.

temperature, wet bulb—equilibrium temperature of water evaporating into air when the latent heat of vaporization is supplied by the sensible heat of the air.

thaw needle—a small tube with valved head, connected by hose to a source of warm water or steam, which may be inserted in the air agitation tube in an ice can for the purpose of thawing the air agitation tube loose from the ice block.

thawing—changing free water, or contained water as in foods, from solid phase to liquid phase by the addition of heat.

thermal conductivity—the ability of a material to transmit heat from one point in the material to another. Indicated in terms of Btu per hr sq ft deg F per in. of thickness.

thermal valve—a valve controlled by a thermally responsive element, for example, a thermostatic expansion valve which is usually responsive to suction or evaporator temperature.

thermocouple—device for measuring temperatures utilizing the effect that an electromotive force is generated whenever two junctions of two dissimilar metals in an electric circuit are at different temperature levels.

thermodynamic properties—basic qualities used in defining the condition of a substance, such as temperature, pressure, volume, enthalpy, entropy.

thermodynamics—the science of heat energy and its transformations to and from other forms of energy.

thermometer—instrument for measuring temperature.

thermostat—automatic control device responsive to temperature.

thermostat, direct-acting—an instrument which permits passage of control air when temperature is higher than the temperature for which instrument is set and cuts off the air when temperature goes below the temperature for which instrument is set.

thermostat, reverse-acting—an instrument which permits passage of control air when temperature is lower than the temperature for which instrument is set and cuts off the air when temperature goes higher than the temperature for which the instrument is set.

throttling—of a fluid, an irreversible adiabatic process which consists of lowering pressure by an expansion without work.

throw—the distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

ton of refrigeration—a rate of heat interchange of 12,000 Btu per hour; 200 Btu per min.

ton-day of refrigeration—the heat removed by a ton of refrigeration operating for a day, 288,000 Btu. Approximately equal to the latent heat of fusion or melting of 1 ton (2,000 lb) of ice.

total cooling effect of a unit—the difference between the total enthalpy of the dry air and water vapor mixture entering the unit per hour and the total enthalpy of the dry air and water vapor (and water) mixture leaving the unit per hour, expressed in Btu per hr.

total heat—obsolescent name for enthalpy.

total tons—the total heat load expressed in tons of refrigeration; the sum of the dry tons and the moisture tons.

transmission—in thermodynamics a general term for heat travel; properly, heat transferred per unit of time.

transmittance—when heat flows from one medium to another, through one or more intermediate resistances, the transmittance of the system is equal to the rate of heat flow per sq ft of a specified side of the boundary, divided by the difference in the temperature of the two surfaces.

tray freezing method—food arranged on shallow trays, placed in portable racks, are placed in insulated chambers and subjected to moving refrigerated air for freezing.

triple point—static point at which three phases of given substance (i.e. solid, liquid and gas) exist in equilibrium.

tubing, soft copper—deoxidized and dehydrated, seamless, soft copper; thoroughly annealed to assure quality for bending, flaring. Commonly used for Freon-12 and methyl chloride.

turbulent flow—fluid flow in which the fluid moves transversely as well as in the direction of the tube or pipe axis, as opposed to streamline or viscous flow.

"*U*" factor—overall coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or other surface for a difference in temperature of 1 deg F between the fluid on the inside and that on the outside of the surface.

unit—1. A factory-made, encased assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, humidifying unit, etc. or 2. Unit, hermetically sealed—refrigerating unit having no exposed mechanical driving connection and containing no shaft seal.

unit cooler—adapted from unit heater to cover any cooling element of condensed physical proportions and large surface, generally equipped with a fan.

unit system—a self-contained system which has been assembled and tested prior to its installation and which is installed without connecting any refrigerant-containing parts. See also self-contained system.

unloader—a device in a compressor for equalizing high and low-side pressures when the compressor stops, and for a brief period after it starts, in order to decrease the starting load on the motor.

usage load—sum of the door load and product load factors.

unit system, removable—a refrigerating system which is readily removable as one unit from the cabinet or space which it cools and from the building in which it is used without disconnecting any refrigerant-containing part of the system.

vacuum system—in refrigeration one which employs a vacuum to boil water at the temperature desired; one which employs evaporating water vapor as the refrigerant.

valve, back-pressure—one controlling flow in the evaporator vapor line because of a fluctuating suction pressure, it is necessary to prevent the evaporator temperature from falling below a pre-determined value.

valve, check—a valve allowing (refrigerant) flow in one direction only.

valve, direct-acting diaphragm—one which closes with the admission of fluid pressure to a diaphragm and opens when pressure is released.

valve, discharge—on a compressor, the valve which allows compressed refrigerant to flow from the cylinder to the discharge main.

valve, expansion—1. Valve for controlling the flow of refrigerant to the cooling element, or 2. Expansion valve, automatic—a device which regulates the flow of refrigerant from the liquid line into the evaporator to maintain a constant evaporator pressure, or 3. Thermostatic—a device to regulate the flow of refrigerant into an evaporator in order to maintain an evaporation temperature in a definite relationship to the temperature of a thermostatic bulb.

valve, emergency relief—a manually-operated valve for the discharge of refrigerant in case of fire or other emergency.

valve, float—1. Regulating valve controlled by liquid level, or 2. Valve actuated by float in a liquid container.

valve, low-side float—a float valve operating by changes in level of low-pressure liquid, opens at low level and closes at high.

valve, king—stop valve between receiver and liquid main.

valve, packless—a valve which does not use packing to prevent leaks around the valve stem. Flexible material usually used to seal against leaks and still permit valve movement.

valve, pressure relief—a valve held closed by a spring or other means, which automatically relieves pressure in excess of its setting.

valve, purge—device to allow fluid to flow out of system, particularly non-condensible gases. Also called a drain valve.

valve, reducing—a valve which maintains a uniform pressure on its outlet side, irrespective of how the pressure on its inlet side may vary above the pressure to be maintained.

valve, reverse-acting diaphragm—opens with the admission of pressure to the diaphragm and closes when pressure is released.

valve, safety—same as pressure relief valve.

valve seat—the stationary portion of the valve which, when in contact with the movable portion, stops flow completely.

valve, stop—a shut-off valve other than a service valve for controlling the flow of the refrigerant.

valve, suction—in a compressor, the valve which allows refrigerant to enter the cylinder from the suction line, and prevents return flow.

vapor—a gas, particularly one near to equilibrium with the liquid phase of the substance. Usually used instead of gas for a refrigerant, and in general for any gases below the critical temperature.

vapor lock—formation of some vapor or all vapor in a liquid line reducing weight flow as compared to weight flow in liquid phase with the same pressure differential.

vapor lock device—an orifice; capillary tube or other device having a restricted passage of fixed size for liquid refrigerant. It restricts flow of vapor of that same liquid to a lower rate of flow (in lb per min) with the same pressure difference.

velocity, critical—the velocity above which fluid flow is turbulent.

ventilation—the process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

vertical bunkers (refrigerated cars and trucks)—filled with ice, the air in the bunker is chilled and flows out of the bottom opening in the bunker into the space beneath the floor rack.

vital heat—heat generated by fruits and vegetables in storage, due to ripening.

volatile liquid—one which evaporates readily at atmospheric pressure and room temperatures.

walk-in box—storage refrigerator.

warm-air duct—in commercial refrigerators the passage through which warm air ascends from storage space to the bunker. In heating, ventilating and air conditioning, pipes or ducts for conveying warm air into or out of a space or room.

water forecooling tank—tank where inlet water is cooled prior to freezing.

water or brine cooler—a factory-made assembly of elements in which the water or brine and the refrigerant are in heat transfer relationship, causing the refrigerant liquid to evaporate and the water or brine to be cooled.

water vapor—used commonly in air conditioning parlance to refer to steam in the atmosphere.

wax—a material which may separate on cooling of oil-refrigerant mixtures. Wax may plug the expansion valve and reduce heat transfer of the coil.

weep—drip from frozen foods.

wet bulb depression—difference between dry bulb and wet bulb temperature.

wet compression—a system of refrigeration in which some liquid refrigerant is mixed with vapor entering the compressor so as to cause the discharge vapors from the compressor to be saturated rather than superheated.

window unit—self-contained room cooler arranged to be supported in or connected with a window opening, circulating outside air over the high side and room air over the low side.

winter air conditioning—heating, humidification, air distribution and air cleaning, where outside temperatures are below the inside or room temperature.

wire drawing—restriction of area for a flowing fluid, causing a loss in pressure by (internal and external) friction without loss of heat or performance of work; throttling.

year 'round air conditioning system—one which ventilates, heats and humidifies in winter and cools and dehumidifies in summer the spaces under consideration, and provides the desired degree of air motion and cleanliness.

42. CODES AND STANDARDS

The Codes and Standards listed below represent accepted practice, methods, or standards prepared and accepted by the organizations indicated. They are valuable guides for the practicing engineering determining test methods, ratings, performance requirements, and limits applying to equipment used in refrigeration and air conditioning. Copies can usually be obtained from the organizations listed in the reference column.

SUBJECT	TITLE	SPONSOR	REFERENCE
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning (1938)	A.S.H.V.E. ASRE	A.S.H.V.E.
Air Conditioning (150,000 Btu/Hr or less)	Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (1945)	N.W.A.H. & A.C.A.	N.W.A.H. & A.C.A. No. 7
Air Conditioning (Above 150,000 Btu/Hr)	The Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (1942)	N.W.A.H. & A.C.A.	N.W.A.H. & A.C.A.
Building Requirements	American Standard Building Requirements (1946)	N.H.A. U.S.P.H.S.	ASA A53.1—1946
Cleaners (Air)	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (1934)	A.S.H.V.E.	See A.S.H.V.E. Transactions Vol. 39, 1933, p. 225
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1945)	H.P. & A.C.N.A.	H.P. & A.C.N.A. Engrg. Stds. Sec. 2 Part V
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1928)	A.S.M.E.	ASA A13—1928
Coils	Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Non-Volatile Refrigerants (1945)	B.C.M.I. B.S.	B.S. T.S. 4044
Compressors	ASRE Standard Methods of Rating and Testing Refrigerant Compressors	ASRE A.S.H.V.E. A.C.R.M.A.	ASRE Standard 23-R
Condensers	ASRE Standard Methods of Rating and Testing Evaporative Condensers	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 20
Condensers	ASRE Standard Methods of Rating and Testing Water-Cooled Refrigerant Condensers	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 22
Condensing Units	ASRE Standard Methods of Rating and Testing Mechanical Condensing Units (1940)	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 14
Condensing Units	Commercial Standard for Commercial Electric Refrigeration Condensing Units (1945)	B.S. S.R.C.A.	B.S. CS107-45
Conductivity	Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative) (1942)	A.S.H.V.E. ASRE A.S.T.M. N.R.C.	A.S.H.V.E.

SUBJECT	TITLE	SPONSOR	REFERENCE
Control Equipment (Industrial)	Underwriter's Laboratories, Inc., Standard for Industrial Control Equipment (July 1938, reprinted May 1950)	U.L.	U.L. Subject 508
Controls	Underwriter's Laboratories, Inc., Standard for Temperature Indicating and Regulating Equipment (Jan 1947)	U.L.	U.L. Subject 873
Convector	Recommended Commercial Standard for Testing and Rating Convectors (1946)	B.S. C.M.A. I.B.R.	B.S. TS-4169
Coolers (Air)	ASRE Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration (1945)	ASRE A.S.H.V.E. A.C.R.M.A. R.E.M.A.	ASRE Standard 25
Coolers	ASRE Standard Methods of Rating and Testing Water and Brine Coolers	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 24
Cooling Units	ASRE Standard Methods of Rating and Testing Air Conditioners (1949)	ASRE A.S.H.V.E. R.M.A. R.E.M.A. A.C.M.A.	ASRE Standard 16-R
Exchangers (Heat)	Standards of Tubular Manufacturers Association (1941)	T.E.M.A.	T.E.M.A.
Fans	Definitions and Terms in Use by the Blower Industry (1946)	N.A.F.M.	N.A.F.M. Bulletin No. 105
Fans	Standard Test Code for Centrifugal and Axial Fans (1938)	N.A.F.M.* A.S.H.V.E.	N.A.F.M. Bulletin No. 103
Fans	N.E.M.A. Standards for Electric Fans	N.E.M.A.	N.A.F.M. Publ. FM 1-1951
Heat Transfer (Walls)	A.S.H.V.E. Standard Test Code for Heat Transmission through Walls (1927)	A.S.H.V.E.	A.S.H.V.E.
Piping	American Standard Code for Pressure Piping (1951)	A.S.M.E.	ASA B31.1—1951
Pumps	Hydraulic Institute Test Code for Centrifugal Pumps. Hydraulic Institute Test Code for Rotary Pumps (1943)	H.I.	H.I. Section F
Refrigeration (Equipment)	Underwriter's Laboratories, Inc., Standard for Air Conditioning and Commercial Refrigerating Equipment (Feb. 1946)	U.L.	U.L. Subject 207A
Refrigeration (Mechanical)	American Standard Safety Code for Mechanical Refrigeration (ASA B9. 1-1950)	ASRE	ASRE Standard 15
Refrigeration (Unit Systems)	Underwriter's Laboratories, Inc., Standard for Unit Refrigerating Systems (Feb. 1946)	U.L.	U.L. Subject 207C

SUBJECT	TITLE	SPONSOR	REFERENCE
Refrigerators (Gas-Fired)	American Standard Approval Requirements for Refrigerators Using Gas Fuel (1941)	A.G.A.	ASA Z21.19—1941
Refrigerators (Household)	American Standard Test Procedure for Household Electric Refrigerators (Mechanically Operated) (1944)	ASRE U.S.D.A.	ASA B38.2—1944 (N.E.M.A. Pub. No. 47-127)
Unit Heaters	American Standard Approval Requirements for Gas Unit Heaters (1940)	A.G.A.	ASA Z21.16—1940
Unfired Pressure Vessels	Unfired Pressure Vessel Code (1950 with 1950 Addenda)	A.S.M.E.	A.S.M.E.
Vacuum Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (1934)	A.S.H.V.E.	A.S.H.V.E.

ABBREVIATIONS

A.C.M.A.[1]	Air Conditioning Manufacturers Association.
A.C.R.M.A.	Air Conditioning and Refrigerating Machinery Association.
A.F.A.	American Foundrymen's Association.
A.G.A.	American Gas Association.
A.G.A.E.M.[2]	Association of Gas Appliance and Equipment Manufacturers.
A.I.L.	Anthracite Industries Laboratory.
ASA	American Standards Association.
A.S. of A.	Acoustical Society of America.
A.S.H.V.E.	American Society of Heating and Ventilating Engineers.
A.S.M.E.	American Society of Mechanical Engineers.
ASRE	American Society of Refrigerating Engineers.
A.S.T.M.	American Society for Testing Materials.
B.S.	National Bureau of Standards.
C.M.A.	Convector Manufacturers Association.
F.H.A.	Federal Housing Administration.
G.A.M.A.	Gas Appliance Manufacturers' Association.
H.I.	Hydraulic Institute.
H.P. & A.C.N.A.	Heating, Piping and Air Conditioning Contractors National Association.
I.B.R.	Institute of Boiler and Radiator Manufacturers Association.
I.C.H.A.M.	Institute of Cooking and Heating Appliance Manufacturers.
I.M.W.I.	Industrial Mineral Wool Institute.
I.U.H.A.	Industrial Unit Heater Association.
N.A.F.M.	National Association of Fan Manufacturers.
N.B.F.U.	National Board of Fire Underwriters.
N.E.M.A.	National Electrical Manufacturers Association.
N.F.P.A.	National Fire Prevention Association.
N.H.A.	National Housing Agency.
N.R.C.	National Research Council.
N.W.A.H. & C.A.	National Warm Air Heating and Air Conditioning Association.
O.B.I.[3]	Oil Burner Institute.
O.H.I.A.	Oil Heat Institute of America.
O.P.A.	Office of Price Administration.
P.F.M.A.	Propeller Fan Manufacturers Association.
R.E.M.A.	Refrigeration Equipment Manufacturers Association.
R.M.A.[1]	Refrigerating Machinery Association.
S.A.E.	Society of Automotive Engineers.
S.B.I.	Steel Boiler Institute.
S.R.C.A.	Standard Refrigeration Compressor Association.
T.E.M.A.	Tubular Exchanger Manufacturers Association.
U.L.	Underwriter's Laboratories.
U.S.D.A.	United States Department of Agriculture.
U.S.P.H.S.	United States Public Health Service.

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[1] Superseded 1940 by A.C.R.M.A.

[2] Superseded 1945 by G.A.M.A.

[3] Superseded 1942 by O.H.I.A.

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REFRIGERATION CLASSIFIED

REFRIGERATING DATA BOOK

Basic Volume

Seventh Edition, 1951

THE "REFRIGERATION CLASSIFIED" Section of the Data Book has been completely revised for this edition. Special care has been taken to augment the listings of materials needed especially for applications pertaining to our defense needs.

The list of headings is intended to include not only complete systems or package units but also all items of materials, components and accessories, which may become, through manufacturing, assembling or installation on the job, integral parts of any type of refrigerating or air conditioning system.

For the convenience of the user, headings are arranged in alphabetical order. The main listings for each item appear under the most generally accepted name for that item. Other names for the same item are found in proper alphabetical order, with cross-references to the main listing. In some instances cross-references are found to other items which may not be the same but which may perform similar functions. In all instances the address of a source of supply is given together with its name.

Advertisers are indicated by bold-face type, followed by a reference to the page number on which their advertising appears.

Although the publishers assume no responsibility for accuracy or completeness of the listings, every effort has been expended to make the listings authentic and comprehensive. Suggested additions, or advice regarding inaccuracies or omissions will be warmly welcomed.

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THE KING ZEERO COMPANY

1447-55 Montrose Ave. — Chicago 13, Illinois

"King Zeero" ICE BUILDERS

for Sweet Water Circulating Systems

Patent No. 2448453

Other Patents Pending

Tons of ice at your disposal without handling.

Extra highly efficient coils for ice building.

Extra highly efficient controlled water circuit for ice melting.

Extra capacity flash cooling.

Complete with 2" corkboard insulation.

Refrigerant float control valve and strainer Safety valve. Oil drain valves. Water gauge column. Hand ice thickness gauge. Extra heavy gauge coils.

Extra heavy gauge tank.

Automatic ice thickness control switch (optional).

Neat appearing. Finished genuine sheet aluminum.

Furnished in 40 models and sizes, both vertical and horizontal, capacities 1,000 to 13,000 lbs. in a single unit.

One or any number of tanks can be connected.

Small floor space required.

Built for long life..

The "KING ZEERO" ice builder connects to your refrigerating system. A circulating pump connects the ice-builder to your secondary equipment. (Milk cooler-vat-dough mixer, etc.)

The ice builder tank is to be almost filled with water. Refrigerant enters the coils in the ice builder through the float valve, or in the ice builder cabinet through the expansion valves. The compressor pulls the evaporated gas from the coils, lowers the coil temperature to a point below freezing and ice forms on the coils. When the ice has a thickness of 2 to 3½ inches the ice-building process can be stopped.

All the water in the ice builder tank is not frozen. When it is desired to use the refrigeration stored up in the ice it is only necessary to start the pump circulating water from the tank through the secondary equipment.

The water absorbs a definite amount of heat in the secondary equipment and returns to the inlet connection on the ice builder tank. From here the warmed water starts a long definite travel through the ice field, gradually losing its heat to the slowly melting ice. At the end of the journey through the ice field the moving water is close to the melting ice temperature. It enters the pump and is sent on its way again through the secondary equipment.

When the day's cooling requirements are completed the melted ice is replaced by simply operating the compressor until the desired thickness is again attained.

Ice is formed on the coils during the off load period. Circulated water during the load period releases the stored up refrigeration as needed.

Will deliver 32° to 34° sweet water for circulation.

Will deliver low temperatures if calcium chloride is added to the water to be frozen.

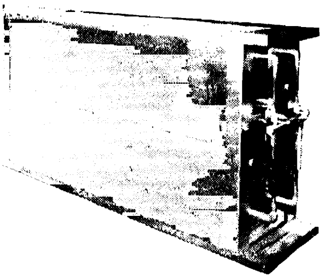
There isn't any handling of the ice formed. Ice is formed and melted on the coils.

A practical way to cool a large volume with a comparatively small refrigerating compressor.

Ammonia, Methyl Chloride or Freon may be used as refrigerant.

A "KING ZEERO" ice builder may be changed from brine to sweet water or vice-versa at any time by simply changing the solution.

CABINET TYPE



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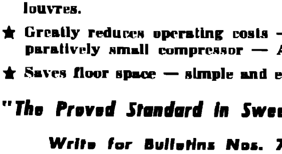

Patents Pending

- ★ **Product Insurance against refrigeration failure.**
- ★ **Non-Mechanical Built-In Agitation.**
- ★ **Combination Flash Cooling and Ice Holdover.**
- ★ **Equipped with "E" type permanently fixed efficiency louvres.**
- ★ **Greatly reduces operating costs — requires only comparatively small compressor — Ammonia, Freon, etc.**
- ★ **Saves floor space — simple and economical to operate.**

"The Proved Standard in Sweet Water Cooling"

Write for Bulletins Nos. 74-E and 95.

Greater Ice Capacity • Lower Cooling Costs • Economical • Efficient

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Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N. J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

ACCESS DOORS

Bahnson Co., 1001 S. Marshall St., Winston-Salem, N.C.
Biersach & Niedermeyer Co., 1937 N. Hubbard St., Milwaukee 12, Wis.
Philip Carey Mfg. Co., Lockland, Cincinnati 15, O.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
York Corp., York, Pa. (*p. 52*)

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ACCUMULATORS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. (*p. 178*)
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Creamery Package Mfg. Co. 1243 W. Washington Blvd., Chicago 7, Ill. (p. 49)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 81*)
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. (*p. 90*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 89*)
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City 7, N.J.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. (*p. 142*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Remco, Inc., Zelienople, Pa. (*p. 87*)
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Temprite Products Corp., E. Maple Rd., Burmingham, Mich. (*p. 89*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.
Wildman Boiler & Tank Co., 3026 Carroll Ave., Chicago 12, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

ACCUMULATORS, ICE STORAGE TYPE

Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. (*p. 152*)
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. (*p. 4*)
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Philadelphia 46, Pa.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
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York Corp., York, Pa. (*p. 52*)

ACTIVATED ALUMINA (See DEHYDRANTS)

ACTIVATED CARBON

Barada & Page, Inc., Guinotte & Michigan Aves., Kansas City 1, Mo.
Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
W. B. Connor Engrg. Corp., Shelter Rock Lane, Danbury, Ct.
Darco Corp., Wilmington, Del.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.

ADAPTER FLANGES

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. (*p. 149*)
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 10[illegible]*)
Mueller Brass Co., Port Huron, Mich.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.

ADHESIVES, CEMENTS, COMPOUNDS

Aetna Plywood & Veneer Co., 1799 Elston Ave., Chicago 22, Ill.
American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Armstrong Cork Co., Lancaster Pa. (*p. 132*)
Atlas Mineral Products Co., Mertztown, Pa.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, O.
Dennis Chemical Co., 2701 Papin St., St. Louis 3, Mo.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Glidden Co., 11001 Madison Ave., Cleveland 2, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Grant Wilson, Inc., 316 S. La Salle St., Chicago 4, Ill.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. (*p. 85*)
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Minnesota Mining & Mfg. Co., 900 Fauquier St., St. Paul 6, Minn.
Miracle Adhesives Corp., 214 E. 53rd St., N.Y.C. 22
Pecora Paint Co., 3501 N. 4th St., Philadelphia 40, Pa.
Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Philadelphia 7, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Plaskon Div., Libbey, Owens, Ford Glass Co., 2112 Sylvan Ave., Toledo 6, O.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Randolph Products Co., Carlstadt, N.J. (*p. 103*)
Stic-Klip Mfg. Co., 50 Regent St., Cambridge 40, Mass. (*p. 127*)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.
Virginia Smelting Co., W. Norfolk, Va. (*p. 35*)

ADSORBENTS (See ACTIVATED CARBON; also DEHYDRANTS)

AERATORS (See WATER COOLERS, BAUDELOT TYPE)

AGITATORS, BRINE, etc.

Arrow Tank Co., Inc., 16 Barnett St., Buffalo 15, N.Y.
Baker Refrigeration Corp., Inc., S. Windham, Me. (*p. 63*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 16, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 89*)
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City 7, N.J.

(Continued)

AGITATORS, BRINE, etc. (Continued)

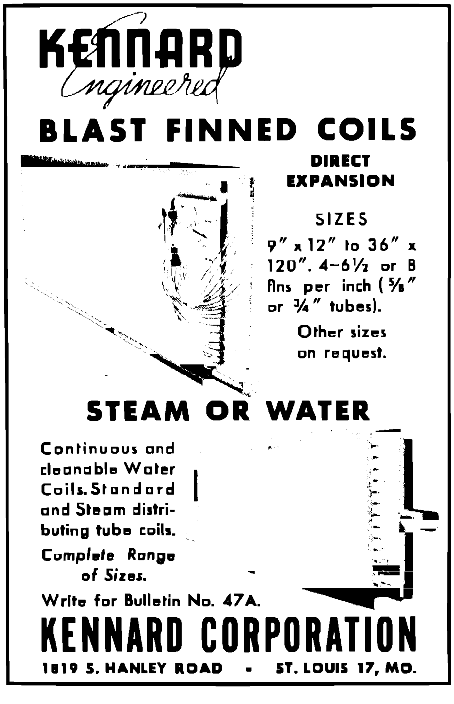
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa.
Nooter Corp., 1426 S. 2nd St., St. Louis 4. Mo.
Piqua Machine & Mfg. Co., Young & College Sts., Piqua, O.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Standard Elec. Mfg. Co., West Berlin, N.J.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

AIR CLEANERS, ELECTRONIC (See also AIR FILTERS)

American Air Filter Co., Inc., 215 Central Ave., Louisville 8, Ky.
Raytheon Mfg. Co., Waltham 54, Mass.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Trion, Inc., 1000 Island Ave., McKees Rocks, Pa.
Westinghouse Elec. Corp., Bloomfield, N.J.

AIR CONDITIONERS, PORTABLE, FLOOR & WINDOWSILL TYPE

Airtemp Div., Chrysler Corp., 1600 Webster St., Dayton 1, O. (*p. 19*)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STEAM OR WATER
Continuous and cleanable Water Coils. Standard and Steam distributing tube coils.
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American Coils Co., 25 Lexington St., Newark 5, N.J.
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y. (*p. 59*)
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
Hastings Air Conditioning Co., Inc., Hastings, Neb.
International Harvester Co., 180 No. Michigan Ave., Chicago 1, Ill.
Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
L. & P. Elec. Co., 684 Bedford Ave., Brooklyn, N.Y.
Mitchell Mfg. Co., 2525 Clybourn Ave., Chicago 14, Ill.
Nevinger Mfg. Co., Greenville, Ill.
Pacific Mfg. Corp., 7500 Stanton Ave., Cleveland 4, O.
Philco Corp., Tioga & C Sts., Phila. 34, Pa.
Pleasantaire Corp., 14th & K Sts., N.W., Washington 5, D.C.
Refrigeration Engrg. Inc., 7250 E. Slauson Ave., Los Angeles 22, Cal. (*p. 196*)
Remington Corp., Willey St., Auburn, N.Y
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 37*)
Westinghouse Elec. Corp., Bloomfield, N.J.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

AIR CONDITIONERS, SELF-CONTAINED

Abbeon Supply Co., 58-10 41st Dr., Woodside, N.Y.
Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
Airtemp. Div., Chrysler Corp., 1600 Webster St. Dayton 1, O. (*p. 19*)
American Coils Co., 25 Lexington St., Newark 5, N.J.
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Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Brunner Mfg. Co., 1821 Broad St., Utica, N.Y. (*p. 48*)
Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
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Frick Co., Waynesboro, Pa. (*p. 51*)
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
L.&.P. Elec. Co., 684 Bedford Ave., Brooklyn, N.Y.
Mitchell Mfg. Co., 2525 Clybourn Ave., Chicago 14, Ill.
Nevinger Mfg. Co., Greenville, Ill.
Refrigeration Engrg. Inc., 7250 E. Slauson Ave., Los Angeles 22, Cal. (*p. 196*)
Servel, Inc., Evansville 20, Ind.
B. F. Sturtevant Div., Westinghouse Elec Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
United Conditioning Corp., 74 Varick St., N.Y.C. 13
U.S. Air Conditioning Corp., Como Ave., S.E. at 33rd St., Minneapolis 14, Minn. (*p. 37*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill)
York Corp., York, Pa. (*p. 52*)

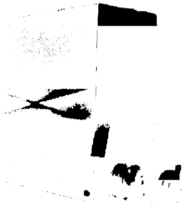
Now More Than Ever

It's Curtis for PROFITS

Because Most Installations are Within the Curtis Range — 2 through 40 Tons

10 and 15 Ton
Complete CENTRAL TYPE

2 through 8 Ton PACKAGED UNITS
(Shipped to you completely assembled)



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Towers and A
Handling Unit

With these units, you can handle any AIR CONDITIONING — REFRIGERATION or AIR MOVING JOB within this range — and when the job is sold, it STAYS SOLD because the QUALITY IS THERE No corners have been cut in the manufacture of Curtis equipment — pound for pound, it's the best buy in the industry

¼ H P through 40 H P
Condensing Units

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TIME

NEWSWEEK

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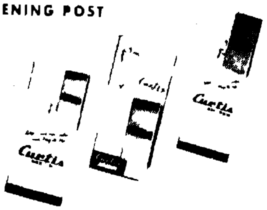

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97 Years of Successful Manufacturing

CURTIS REFRIGERATING MACHINE DIVISION
of Curtis Manufacturing Company
1949 Kienlen Ave St Louis 20 Mo

Curtis

R 51 1A

AIR CONDITIONING COILS, CHILLED WATER

(C—also with cleanable tubes)

(C) Aerofin Corp., 410 Geddes St., Syracuse, N.Y. (*p. 11*)
American Coils Co., 25 Lexington St., Newark 5, N.J.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
(C) Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
(C) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. (*p. 40*)
(C) Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Conditionaire Unit Co., Div. of Howe Ice Machine Co., 2815 Montrose Ave., Chicago 18, Ill. (*p. 198*)
(C) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles, 33, Cal.
C. A. Dunham Co., 400 W. Madison St., Chicago 6, Ill.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
(C) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
(C) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(C) Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa. (*p. 57*)
(C) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. (*p. 6*)
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
(C) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
(C) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. (*p. 200*)
Modine Mfg. Co., Racine, Wis.
(C) John J. Nesbitt, Inc., Phila. 36, Pa. (*p. 9*)
Nevinger Mfg. Co., Greenville, Ill.
(A, B) Pace Co., 116 N.W. Couch, Portland, Ore.
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. (*p. 201*)
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(C) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)
V. E. Sprouse Co., Inc., Columbus, Ind.
(C) B. F. Sturtevant Div., Westinghouse Elec. Corp. 101 Readville St., Boston 36, Mass. (*p. 23*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles, Cal.
(C) Super Radiator Corp., 6714 Walker St., St. Louis Park, Minneapolis 16, Minn.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(C) Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
(C) U.S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 37*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
(C) Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)
(C) Young Radiator Co., Racine, Wis. (*p. 77*)

AIR CONDITIONING COILS, DIRECT EXPANSION

Aerofin Corp., 410 Geddes St., Syracues, N.Y. (*p. 11*)
American Coils Co., 25 Lexington St., Newark 5, N.J.
Baker Refrigeration Corp., S. Windham, Me (*p. 63*)
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Betz Corp., 445 State St., Hammond, Ind.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Bush Mfg., 179 South St., W. Hartford 10, Ct. (*p. 40*)
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Conditionaire Unit Co., Div. of Howe Ice Machine Co., 2815 Montrose Ave., Chicago 18, Ill. (*p. 198*)
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
C. A. Dunham Co., 400 W. Madison St., Chicago 6, Ill.
Frick Co., Waynesboro, Pa. (*p. 51*)
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 198*)
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. (*p. 6*)
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. (*p. 41*)
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. (*p. 200*)
Modine Mfg. Co., Racine, Wis.
Nevinger Mfg. Co., Greenville, Ill.
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. (*p. 201*)
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4 O,
V. E. Sprouse Co., Inc., 1804 22nd St., Columbus, Ind.
B. F. Sturtevant Div., Westinghouse Elec. Corp.) 101 Readville St., (*p. 23*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Super Radiator Corp., 6714 Walker St., St. Louis Park, Minneapolis 16, Minn.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
U.S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 37*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)
Young Radiator Co., Racine, Wis. (*p. 77*)

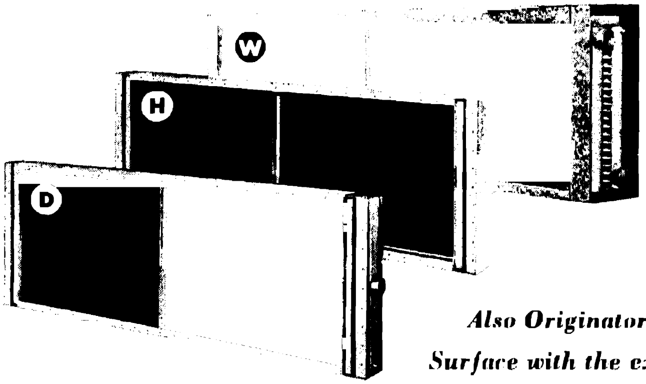
AIR CONDITIONING COILS, HEATING, HOT WATER

Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *p. (11)*
American Coils Co., 25 Lexington St., Newark 5, N.Y.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. (*p. 40*)
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. (*p. 7*)
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.

(Continued)

NESBITT SURFACE

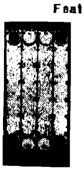
FOR HEATING, DRYING, COOLING, DEHUMIDIFYING



Originators of FREEZEPROOF Heating Surface with Dual Steam-Distributing Tubes

Also Originators of Cooling (Water) Surface with the exclusive drain feature —and with the Surface pitched in the casing!

Exclusive Nesbitt Drainability Feature



Cleanable Water Surface with Removable Headers

SERIES W Continuous tube or cleanable tube water surface for air-cooling and dehumidifying with cold water, or air heating with hot water. Constructed of copper tubes and plate-type aluminum fins. Single, double and half-serpentine arrangements. Available in wide range of sizes in three types: TYPE WD sections incorporate the exclusive Nesbitt drainability feature (illustrated at left) and have the surface pitched in the casing. Positive drainage of all tubes and return bends is insured, protecting the surface against winter freeze-ups. Publication 246. TYPE WB sections are for booster-heating or air-cooling applications where air volumes are relatively small and drainability unnecessary. Constructed same as Type WD sections except that the surface is set level in the casing and the drainability feature is omitted. Publication 246. TYPE WC sections employ standard Nesbitt Series W Surface cores, pitched in the casing, and are designed for applications where the tubes require periodic cleaning. Headers are constructed of close-grained gray cast iron with removable cover plates (illustrated at left) permitting inspection of the interior without disturbing connection piping. Drainability is obtained by removing drain tappings at both ends of the section. Available in single and double serpentine circuits in a range of sizes. Publication 255.

SERIES H Highly efficient blast-coil heating surface designed for general heating, ventilating, air-conditioning and drying in both high- and low-pressure steam systems. Copper tubes and headers, aluminum fins, lightweight like Series D Surface, but without steam-distributing tubes. Available in seven surface types, full range of sizes. Publication 248.

Nesbitt Surface is manufactured and sold by

JOHN J. NESBITT, INC.

Branch Offices in Principal Cities

SERIES D Heating surface with Steam-Distributing Tubes. Freezeproof *plus*. Even distribution of the smallest amount of steam over the full length of the surface; therefore: UNIFORM discharge temperatures; perfect controllability with modulating valves; inherent protection against freezing. Ideally suited for preheating outdoor air. Copper tubes and headers, aluminum fins. Available in Type DS, with single or DUAL steam-distributing tubes, single steam headers, in lengths up to six feet; and Type DD, with DUAL steam-distributing tubes, two supply headers for greater lengths up to ten feet. Publication 247.

WRITE TO JOHN J. NESBITT, INC., PHILADELPHIA 36, PA., FOR COMPLETE CATALOGS AND DATA.

AIR CONDITIONING COILS, HEATING, HOT WATER (Continued)

General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
Industrial Mfg. & Engrg. Co., 3846 N. Ravenswood Ave., Chicago 13, Ill.
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. *(p. 6)*
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 195)*
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
Modine Mfg. Co., Racine, Wis.
John J. Nesbitt, Inc., Phila. 36, Pa. *(p 9)*
Nevinger Mfg. Co., Greenville, Ill.
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 105)*
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
V. E. Sprouse Co., Inc., Columbus, Ind.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 23)*
Super-Cold Corp., 1020 E. 59th St., Los Angeles, Cal.
Super Radiator Corp., 6714 Walker St., St. Louis Park, Minneapolis 16, Minn.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Trane Co., La Crosse, Wis. *(p. 118)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y
U. S. Air Conditioning Corp., Como Ave., S.E. at 33rd St., Minneapolis 14, Minn. *(p. 37)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*
Young Radiator Co., Racine, Wis. *(p. 77)*

AIR CONDITIONING COILS, HEATING, STEAM

(F—Freeze Proof Construction available)

(F) Aerofin Corp., 410 Geddes St., Syracuse, N.Y *(p. 11)*
American Coils Co., 25 Lexington St., Newark 5, N.J.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
(F) Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
(F) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
(F) Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. *(p. 7)*
(F) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(F) G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
(F) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
(F) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
(F) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. *(p. 6)*
(F) King Co., 902 N. Cedar St., Owatonna, Minn.
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p.195)*
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(F) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(F) Marlo Coil Co., 6135 Manchester Ave., St. Louis, 10, Mo. *(p. 200)*
Modine Mfg. Co., Racine, Wis.
(F) John J. Nesbitt, Inc., Phila. 36, Pa. *(p. 9)*
Nevinger Mfg. Co., Greenville, Ill.
Refrigeration Appliances, Inc., 918 W. Lake St., Chicago 7, Ill. *(p. 201)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(F) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 105)*
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O
V. E. Sprouse Co., Inc., Columbus, Ind.
(F) B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 23)*
(F) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Super Radiator Corp., 6714 Walker St. St. Louis Park, Minneapolis 16, Minn.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(F)Trane Co., La Crosse, Wis. *(p. 118)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
(F) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
(F) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*
(F) Young Radiator Co., Racine, Wis. *(p. 77)*

AIR CONDITIONING FAN COIL UNITS

(C—Ceiling mounted; F—Floor mounted)

Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. *(p. 19)*
(F) American Coils Co., 25 Lexington, St., Newark 5, N.J.
(C,F) Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*

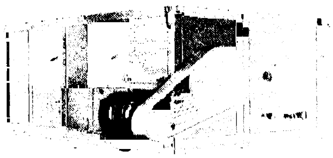
(Continued)

KENNARD *Engineered*

AIR CONDITIONING BLOWER UNITS

COMFORT OR INDUSTRIAL

Freon — Water — Ammonia

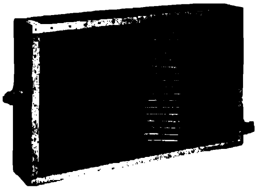


Sizes 300 to 20,000 CFM.
Vertical and Horizontal types.

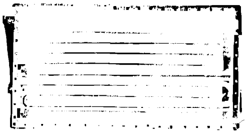
Write for Bulletin No. 486.

OHA Unit "Space Saver" for office, hotel and apartment applications. ½ to 1½ tons.

KENNARD CORPORATION
1819 S. HANLEY ROAD • ST. LOUIS 17, MO.



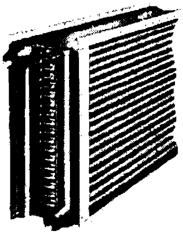
Flexitube Aerofin



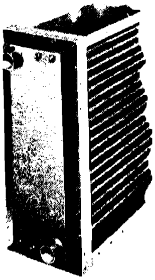
Universal Aerofin

AEROFIN

Heat-Exchange Surface



Aerofin Continuous Tube Water Coil

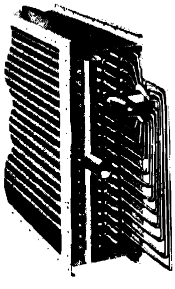


Cleanable Tube Unit with Removable Header

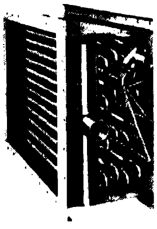
PROVEN PERFORMANCE

- **Aerofin fan-system heat-exchange surface is adaptable to meet all practical requirements of refrigeration. The line is complete—for direct-expansion refrigerants and water.**

- **An unequalled background of development, research and manufacturing is assurance of operating efficiency, long service life and low maintenance costs. Aerofin performance is *proven performance*.**



Direct Expansion Unit with Centrifugal Header



Aerofin Direct Expansion Unit

- **Send for technical literature or consult your Aerofin district office for the solution of your heat-exchange problem.**

AEROFIN CORPORATION

410 South Geddes St., SYRACUSE 1, N. Y.

NEW YORK • CHICAGO • CLEVELAND • DETROIT
PHILADELPHIA • DALLAS • SAN FRANCISCO • TORONTO • MONTREAL

AIR CONDITIONING FAN COIL UNITS (Continued)

(C)Betz Corp., 445 State St., Hammond, Ind.
(C,F) Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
(C,F) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
(C,F) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(C,F) Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
(C) Conditionaire Unit Co., Div. of Howe Ice Machine Co., 2815 Montrose Ave., Chicago 18, Ill. *(p. 198)*
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. *(p. 7)*
(C,F) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(F) Frick Co., Waynesboro, Pa. *(p. 51)*
(C) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(C,F) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
(C,F) Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
(C,F) Hastings Air Conditioning Co., Inc., Hastings, Neb.
(C,F) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 198)*
(C,F) Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
(C,F) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. *(p. 10)*
(C) King Co., 902 N. Cedar St., Owatonna, Minn.
(C) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 195)*
(F) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(C,F) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(C,F) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
(C,F) Merchant & Evans Co., 2035 Washington Ave., Phila. 46, Pa.
(C,F) Modine Mfg. Co., Racine, Wis.
(C,F) John J. Nesbitt, Inc., Phila. 36, Pa. *(p. 9)*
(C,F) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
(C,F) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
(C,F) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
(C,F) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(C) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(C,F) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill
(C,F) B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 28)*
(C,F) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(C) Tenney Engrg., Inc., 26 Ave B, Newark 5, N.J.
(C,F) Trane Co., La Crosse, Wis. *(p. 118)*
(C,F) Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
(C,F) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
(C,F) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Wilson Engrg. Corp., 122 S. Michigan Ave., Chicago 3, Ill.
(C,F) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(C,F) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(C,F) York Corp., York, Pa. *(p. 52)*
(C,F) Young Radiator Co., Racine, Wis. *(p. 77)*

AIR DIFFUSERS (See also GRILLES)

A-J Mfg. Co., 2119 Washington, Kansas City 8, Mo.
Air Control Products, Inc., Coopersville, Mich.
Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Anemostat Corp. of America, 10 E. 39th St., N.Y.C. 16 *(p. 14)*
Barber-Colman Co., Rockford, Ill
W. B. Connor Engrg. Corp., Shelter Rock Lane, Danbury, Ct.
Chas. Demuth & Sons, Inc., 245 Elm Place, Mineola, N.Y.
Excelsior Steel Furnace Co., 118 S. Clinton, Chicago 6, Ill.
Hart & Cooley Mfg Co., 500 E. 8th St., Holland, Mich.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Independent Register Co., 3747 E. 93rd St., Cleveland 5, O.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Pyle-Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
V. E. Sprouse Co., Inc., Columbus, Ind.
Titus Mfg. Corp., Waterloo, Ia.
Tuttle & Bailey, Inc., New Britain, Ct.
U.S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*

AIR DISTRIBUTION PANELS

Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Pyle-Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill

AIR FILTERS (See also AIR CLEANERS)

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
Air Filter Corp., 108 N. Water St., Milwaukee 2, Wis.
Air-Maze Corp., 5200 Harvard Ave., Cleveland 5, O.
Air Stream Filter Corp., 2100 Washington Ave., St. Louis 3, Mo.
American Air Filter Co., Inc., 215 Central Ave., Louisville 8, Ky.
A. G. Brauer Supply Co., 2100 Washington Ave., St. Louis 3, Mo.
Conoflow Corp., 2100 Arch St., Phila. 3, Pa.
Continental Air Filters, Inc., 2520 Helm St., Louisville 1, Ky.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.
Excelsior Steel Furnace Co., 118 S. Clinton, Chicago 6, Ill.
Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Filtair Mfg. Co., 487 S. Van Ness Ave., San Francisco 3, Cal.
Glasfloss Corp., 155 E. 44th St., N.Y.C. 17
Hugo Mfg. Co., 49 Ave., W., & Superior St., Duluth 7, Minn.
Logan Engrg. Co., 4801 W. Lawrence, Chicago 30, Ill.
National Air Conditioning, Inc., Johnstown, Pa.
Northern Blower Co., W. 65th St., South of Denison, Cleveland 2, O.
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 13)*
Research Products Corp., 1015 E. Washington Ave., Madison 3, Wis.
H. J. Somers, Inc., 6063 Wabash Ave., Detroit 8, Mich.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 28)*
Trion, Inc., 1000 Island Ave., McKees Rocks, Pa.
Universal Air Filter Corp., 332 W. Michigan St., Duluth 2, Minn.
Wilson & Co., Inc., 4100 S. Ashland Ave., Chicago 9, Ill.

AIR PURIFICATION EQUIPMENT AND MATERIAL (See also LAMPS, BACTERICIDAL)

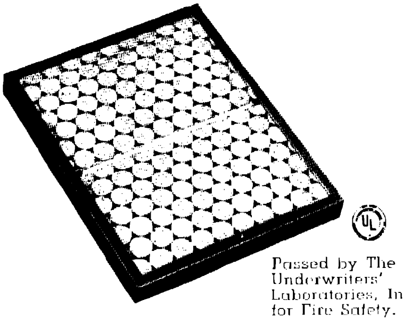
Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
Airkem, Inc., 241 E. 44th St., N.Y.C. 17
American Solvent Recovery Corp., 8th Ave. & Cassady Rd., Columbus 3, O.
W. B. Connor Engrg. Corp., Shelter Rock Lane, Danbury Ct.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
National Air Conditioning, Inc., Johnstown, Pa.
Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Reynolds Elec. Co., 3000 River Rd., River Grove, Ill.
O. A. Sutton Corp., 1812 West 2nd St., Wichita, Kan.

DUST STOP*

T. M. REG. U. S. PAT. OFF.

AIR FILTERS

—a OWENS-CORNING FIBERGLAS product



Passed by The Underwriters' Laboratories, Inc., for Fire Safety.

For Large Heating, Ventilating and Air-Conditioning Systems

DUST-STOP replaceable-type air filters effectively and economically cleanse air of dust, dirt and lint. They are used wherever air is moved mechanically, and to protect machinery and equipment against abrasive dusts.

DUST-STOP Air Filters are made of packs of glass fibers coated with adhesive, faced with metal grills, framed in fiberboard. The glass fibers are inorganic, chemically stable, and resistant to heat, corrosive vapors and most acids. The packs are covered with a non-odorous, non-evaporating adhesive coating—a coating which is not taken up by the non-absorbent glass fibers. Each impinged dust particle is quickly soaked, acting as a wick to carry adhesive to other particles.

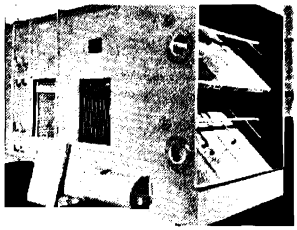
The efficiency, low cost, ready availability and easy replacement of these standard-sized filters have made them standard equipment on leading makes of air-conditioning and heating equipment.

Conventional "L" Type Frame

"V" Type Frame for Restricted Frontal area

In most residential forced-warm-air furnaces and air-conditioners, Dust-Stops are standard equipment.

Dust-Stop Filters are engineered into package air-conditioning units used in commercial establishments.



Fiberglas Aeration Packs are used as contact and eliminator mats in air-washing and humidity control units.

**DUST-STOP is the trade-mark of Owens-Corning Fiberglas Corporation for impingement type air filters made of glass fibers*

For Complete Information See Sweet's Files or Write:

OWENS-CORNING FIBERGLAS CORPORATION

2012 Nicholas Bldg., Toledo 1, Ohio

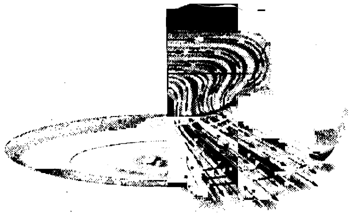
Offices: Atlanta, Baltimore, Boston, Buffalo, Chicago, Cincinnati, Cleveland, Columbus, Dallas, Dayton, Denver, Detroit, Grand Rapids, Houston, Kansas City, Louisville, Los Angeles, Milwaukee, Minneapolis, New York, Philadelphia, Pittsburgh, Portland, Richmond, St. Louis, San Francisco, Seattle, Toledo, Tulsa, Washington, D. C.

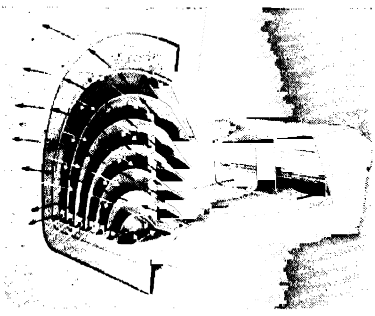
Anemostat Corporation of America

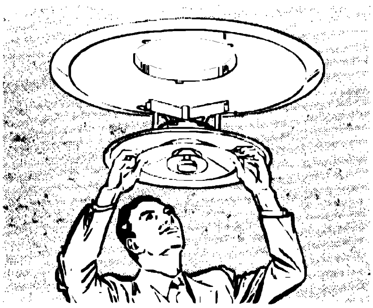
10 East 39th Street
New York 16, N.Y.

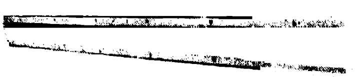
ANEMOSTAT
DRAFTLESS *Aspirating* AIR DIFFUSERS

Representatives in
Principal Cities









PRINCIPAL OF ASPIRATION

The Anemostat Air Diffuser splits the supply air into multiple, separate air streams and at the same time creates low pressure areas in certain parts of diverging passageways. This causes room air to be siphoned into the device where it is thoroughly mixed with supply air. The mixture is discharged in a series of expanding turbulent air layers and, therefore, readily entrains a large amount of additional room air.

As a result, temperature and humidity are equalized throughout the room and a continuous air motion in the occupancy zone within low acceptable velocity limits is established. Stagnant air pockets are also avoided.

The quantity of room air drawn into Anemostat Air Diffusers depends on the specific design, size and type of the various units and is equal to as much as 35 per cent of the supply air. This effective aspiration distinguishes Anemostat Air Diffusers from all other air outlets.

SMALLER DUCTS

Aspiration permits the use of higher temperature differentials and higher supply air velocities than customary, which results in savings in the initial cost of fans and ducts and in the operating cost of fans. Duct layouts may be simplified because Anemostat Air Diffusers distribute air evenly in spite of unusual room plans, columns or other obstructions.

THE COMPLETE LINE

Anemostat Corporation of America has developed air distribution devices and related equipment for every purpose. Round, semi-circular, square and straightline Anemostat Air Diffusers of various types and sizes for ceiling and wall distribution are available for comfort conditioning and industrial heating, ventilating and air conditioning installations.

QUALITY

Anemostat Air Diffusers are scientifically designed according to modern fluid flow theory and manufactured according to modern production standards.

ANEMOSTAT PATENTS

The Anemostat Corporation of America holds over fifty patents covering the use and design of diffusers in the field of air distribution. The superior functioning of the Anemostat is protected by these patents which are the result of inventive spirit, diligent research and constructive engineering.

RESEARCH AND ENGINEERING

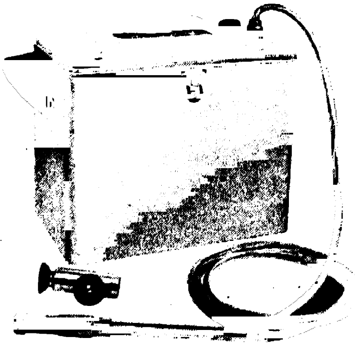
Anemostat Corporation of America maintains a large, well equipped laboratory manned by experienced scientific and technical personnel for testing purposes and the development of new products. Engineers from all over the world have visited the Anemostat laboratory to witness demonstrations of Anemostat products and their applications to any conceivable problem pertaining to air diffusion. Anemostat has spent over a million dollars in research and engineering and will continue to make substantial contributions to the progress of the science of air distribution.

EXPERIENCE

Over a million Anemostat Air Diffusers are now doing an excellent job in thousands of comfort conditioning and industrial heating, ventilating and air conditioning installations. In addition to applications in commercial buildings, industrial plants, hotels, stores, hospitals, theatres, restaurants and homes, Anemostat Air Diffusers are used in military and commercial aircraft, railroad cars, buses and ships.

SPECIAL INSTRUMENTATION

Anemostat engineers have developed various instruments for the accurate and convenient testing of the performance of air outlets. The Anemotherm Air Meter, a self-contanied, compact, portable instrument, facilitates the balancing and checking of air distribution systems by engineers and contractors. It is now commercially available and is being used with great success by engineers and contractors throughout the country.



AIR WASHERS & SPRAY TYPE CONDITIONERS

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
Airwasher Corp., Eaton Rapids, Mich.
American Blower Corp., Div. of American Radiator Standard Sanitary Corp., 8111 Tireman Ave., Detroit, 32, Mich.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Baker Refrigeration Corp. S. Windham, Me. *(p. 63)*
Bayley Blower Co., 66th & Burnham Sts., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E. Cleveland 14, O.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Comfort Products Corp., 2220 S. La Mesa, Dallas 2, Tex.
Continental Air Filters, Inc., 2520 Helm St., Louisville 1, Ky.
Drying System, Inc., 1810½ Foster Ave., Chicago 40, Ill.
Freshn'd-Aire Co., Div. of Corey Corp., 221 N. La Salle St., Chicago 1, Ill.
General Electric Co., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
International Metal Products Co., 500 S. 15th St., Phoenix, Ariz.
Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 260)*
D. J. Murray Mfg. Co., 1024-3rd St., Wausau, Wis.
National Air Conditioning, Inc., Johnstown, Pa.
National Engrg. & Mfg. Co., 519 Wyandotte, Kansas City 6, Mo.
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville, Boston 36, Mass. *(p. 23)*
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. *(p. 118)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
York Corp., York, Pa. *(p. 52)*

ALARM BELLS (See SIGNALS)

ALCOHOL

Barada & Page, Inc., Guinotte & Michigan Aves., Kansas City 1, Mo.
Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.

ALLOYS, BRAZING

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
All-State Welding Alloys Co., Inc., 273 Ferris Ave., White Plains, N.J.
American Platinum Wks., 231 New Jersey R.R. Ave., Newark, N.J.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Handy & Harman, 82 Fulton St., N.Y.C. 7
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City, 7, N.J.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

ALLOYS, DIE CASTING

Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Federated Metals Div., American Smelting & Refining Co., 120 Broadway, N.Y.C. 5
National Lead Co., 111 Broadway, N.Y.C. 6
New Jersey Zinc Co., 160 Front St., N.Y.C. 7.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.

ALLOYS, ELECTRICAL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Armco Steel Corp., Middletown, O.
Carpenter Steel Co., Reading, Pa.
Driver-Harris Co., Harrison, N.J.
Graphite Metallizing Corp., 1013 Nepperhan Ave., Yonkers 3, N.Y.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
United States Steel Co., 525 Wm Penn Place Bldg., Pittsburgh 30, Pa.

ALLOYS, GLASS SEALING

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Carpenter Steel Co., Reading, Pa.
Driver-Harris Co., Harrison, N.J.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

ALUMINUM (See particular mill forms, i.e., BAR, SHEET. etc.)

AMMETERS (See METERS, ELECTRIC)

AMMONIA, ANHYDROUS & AQUA

Armour & Co., 1355 W. 31st St., Chicago 9, Ill.
Barada & Page, Inc., Guinotte & Michigan Aves., Kansas City 1, Mo.
Barrett Div., Allied Chemical & Dye Corp., 40 Rector St., N.Y.C. 6
Henry Bower Chemical Mfg Co., Gray's Ferry Rd. & 29th St., Phila. 46, Pa.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Wilmington 98, Del.
Mathieson Chemical Corp., Mathieson Bldg., Baltimore 3, Md.
Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Phila 7, Pa.

AMMONIA MASKS (See GAS MASKS)

AMMONIA SEPARATORS

Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Cold Control, Inc., 111 Broadway, N.Y.C. 6
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Frick Co., Waynesboro, Pa. *(p. 51)*
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
XL Refrigerating Co., 1834 W. 59th St., Chicago 6, Ill.

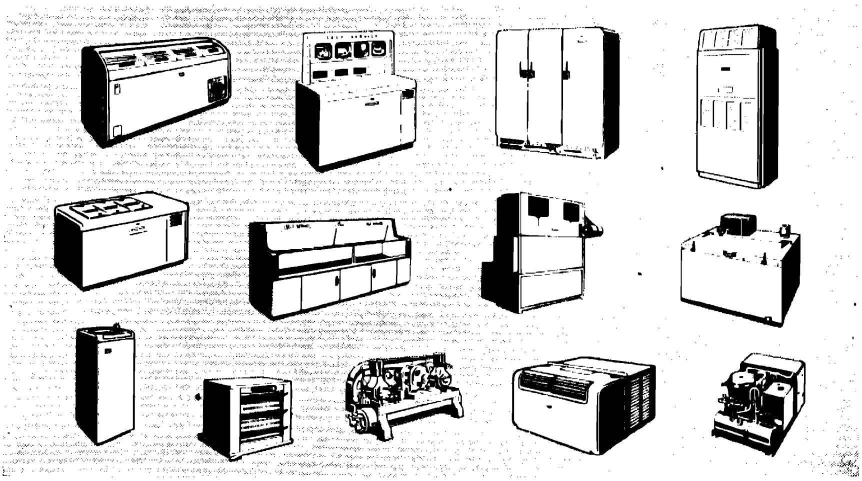
ANCHORS & SHIELDS, EXPANSION, etc.

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6

(Continued)

FRIGIDAIRE

...most complete line in the industry!



Over 400 commercial and air conditioning products—plus the popular line of home appliances—bear the famous Frigidaire emblem . . . the sign of the leader.

Yes, Frigidaire not only leads in the number and type of products—it has also pioneered many of the important advances in refrigeration over the last 30 years.

Frigidaire Commercial Refrigeration Equipment includes: Compressors . . . Cooling Units . . . Reach-In Refrigerators . . . Display Cases . . . Ice Cream Cabinets . . . Zero Self-Servers . . . Beverage Coolers . . . Milk Coolers . . . Water Coolers . . . Ice Makers . . . Evaporative Condensers.

Frigidaire Air Conditioning Equipment includes: Room Air Conditioners . . . Self-Contained Air Conditioners . . . Heavy-Duty Self-Contained Air Conditioners . . . High Capacity Central System Air Conditioning.

Frigidaire Home Appliances include: Refrigerators . . . Food Freezers for Home and Farm . . . Electric Ranges . . . Laundry Equipment consisting of Automatic Washer. Automatic Electric Clothes Dryers, Electric Ironers . . . Electric Water Heaters (both Table-Top and Round Upright models) . . . Kitchen Cabinets and Sinks . . . Electric Dehumidifier.

Frigidaire reserves the right to change specifications, or discontinue models, without notice.

Frigidaire Division of General Motors Corporation

Dayton 1, Ohio • In Canada, Leaside 12, Ontario

ANCHORS & SHIELDS (Continued)

Paine Co., 2951 Carroll Ave., Chicago 12, Ill
Rawlplug Co., Inc., 271 Church St., N.Y.C. 17
Scoville Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Star Expansion Bolt Co., 147 Cedar St., N.Y.C. 6
U. S. Expansion Bolt Co., York, Pa.

ANGLES (See SHAPES, STRUCTURAL)

ANHYDROUS AMMONIA (See AMMONIA)

ANODES, PLATING

American Brass Co., Waterbury 20, Ct. (*p. 189*)
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98 Del.
Federated Metals Div. American Smelting & Refining Co., 120 Bway., N.Y.C. 5
Handy and Harman, 82 Fulton St., N.Y.C. 7
Hanson-Van Winkle Munning Co., Matawan, N.J.
Illinois Zinc Co., 2959 W. 47th St., Chicago 32, Ill.
New Haven Copper Co., Seymour, Ct. (Copper)
New Jersey Zinc Co., 160 Front St., N.Y.C. 7
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
United Chromium, Inc., 100 E. 42nd St., N.Y.C. 17

ASBESTOS (See INSULATION, ASBESTOS, and mill forms, i.e., SHEET, etc.)

ASPHALT (See COATINGS)

ASSEMBLIES, STAMPED, WELDED, etc. (See STAMPINGS)

ATMOSPHERIC COOLERS & ACCESSORIES

Air Equip. Co., 2730 Zuni St., Denver, Col.
Aqua-Mist Co., 426 Jefferson St., Topeka, Kan.
Thomas Beckett & Co., Inc., P.O. Box 7354, Dallas 2, Tex.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Central States Mfg. Co., Inc., 1200 S. Summit St., Arkansas City, Kan.
Comfort Products Corp., 2220 S. La Mesa, Dallas 2, Tex.
Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
International Metal Products Co., 500 S. 15th St., Phoenix, Ariz.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. (*p. 41*)
Lehigh Fan & Blower Co., 519 Wyandotte, 128 Linden St., Allentown, Pa.
National Air Conditioning, Inc., Johnstown, Pa.
National Engrg. & Mfg. Co.,Kansas City 6, Mo.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
Nooter Corp. 1426 S. 2nd St., St. Louis 4, Mo.
Palmer Mfg. Corp., 2200 W. Fillmore, Phoenix, Ariz.
Polaire Cooler Co., Inc., 141 E. Jackson St., Phoenix, Ariz.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Rheem Mfg. Co., 570 Lexington Ave., N.Y.C. 22
Rhode Island Humidifier & Ventilating Co., 99 Chauncey St., Boston 11, Mass.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St Louis 6, Mo.
Simplex Mfg. Co., 1135-3rd St., Oakland 20, Cal.
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N. J.
Thermador Elec'l. Mfg. Co., 5119 District Blvd., Los Angeles, Cal.
Trane Co., La Crosse, Wis. (*p. 118*)
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 37*)
Utility Appliance Corp., 4851 S. Alameda St., Los Angeles 11, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

ATOMIZERS, HUMIDIFYING (See NOZZLES)

AUTOMOBILE AIR CONDITIONING SYSTEMS

Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. (*p. 152*)
Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.
Keco Industries, Inc., 2519 Gilbert Ave., Cincinnati 6, O.

BABBIT METAL (See BEARING MATERIAL)

BACK PRESSURE, etc. (See SUCTION PRESSURE)

BACTERICIDAL LAMPS (See LAMPS, BACTERICIDAL)

BAFFLES (See COIL & BAFFLE ASSEMBLIES)

BAKERY REFRIGERATORS (See BREAD COOLERS; also DOUGH RETARDERS; also FERMENTATION ROOMS)

BALL JOINTS (See SWING JOINTS)

BALLS, FLOAT (See FLOAT BALLS)

BALLS, STEEL BEARING

Carpenter Steel Co., Reading, Pa.
New Departure Div., Gen'l. Motors Corp., Bristol, Ct.
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa

BAR & ROD, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Belmont Smelting & Refining Wks., Inc., 330 Belmon Ave., Brooklyn 7, N.Y.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Mueller Brass Co., Port Huron, Mich.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.

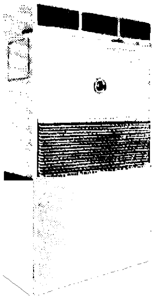
BAR & ROD, BRASS, BRONZE, COPPER, etc.

American Brass Co., Waterbury 88, Ct. (*p. 189*)
Bristol Brass Corp., Bristol, Ct.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
C. G. Hussey & Co., 2860-2nd Ave., Pittsburgh 19, Pa.
Lewin-Mathes Co., 1111 Chouteau, St. Louis, Mo.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Scoville Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Titan Metal Mfg. Co., Bellefonte, Pa.

BAR & ROD, NICKEL & NICKEL ALLOY

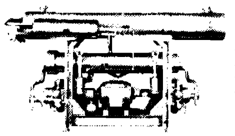
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
International Nickel Co., 67 Wall St., N.Y.C.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

• "PACKAGED" AIR CONDITIONERS
• CENTRAL SYSTEM COOLING EQUIPMENT
• INDUSTRIAL REFRIGERATION UNITS

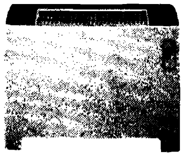


"PACKAGED" AIR CONDITIONERS 2-, 3-, 5-, 8- and 11-ton capacities. Completely assembled and tested at the factory. Cool, dehumidify, filter and circulate air. Free air discharge or duct distribution. Heating coil for year-'round service optional. COMPACT—Occupy very little floor space. EASILY INSTALLED—Need only three connections. FLEXIBLE—Can be installed singly or in multiple. SEALED RADIAL COMPRESSOR—Quiet with all moving parts balanced and bathed in oil for long life. ROOM AIR CONDITIONERS which fit right in window cool, filter, ventilate and circulate air available in ¾ hp and 1 hp sizes.

"PACKAGED" LIQUID CHILLERS



Completely assembled and tested at factory, these Liquid Chiller assemblies furnish chilled water or brine under controlled temperatures for large Air Conditioning and Refrigeration installations. Individual unit capacities up to 200 tons.

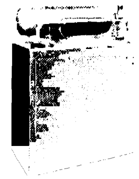


INDIVIDUAL ROOM AIR CONDITIONING UNITS Used for year-'round cooling and heating by connecting with central chilled and hot water systems. Seven models in three types—floor, wall and ceiling—are available. Individual unit control from built-in thermostat. Two centrifugal fans provide air circulation.

DEHUMIDIFIERS ⅛- and ⅕-hp Capacities. Low-cost, portable dehumidifier removes excessive moisture from the air. Can be plugged into any convenient electrical outlet. Protects against rust, mildew and corrosion. Very compact in size—⅕-hp is 34 in. high—16 in. diameter. ⅛-hp is 15 in. high, 12 in. wide, 16½ in. long.

RADIAL COMPRESSOR AIR CONDENSING UNITS 10- to 100-ton capacities. Heavy-duty units, for use with Freon 12 and 22, are especially designed for refrigeration and air conditioning. Radial compressors are direct connected and have force-feed lubrication. Automatic capacity-reduction device. Light weight, economical to operate.

MACHINE TOOL COOLANT COOLERS



Keep coolants and machine tool cutting oils at a constant temperature. This provides greater precision and higher production plus longer tool life. One of the units shown can serve several small machine tools.

Chrysler CHRYSLER AIRTEMP *Airtemp*

AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON 1, OHIO

IN CANADA: THERM-O-RITE PRODUCTS, LTD., TORONTO

BAR & ROD, STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Armco Steel Corp., Middletown, O.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Henry Disston & Sons, Inc., Tacony, Phila. 35, Pa.
Firth-Sterling Steel & Carbide Corp., Demmler Rd., McKeesport, Pa.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 10, Mo.
McInness Steel Co., Corry, Pa.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Weirton Steel Co., Weirton, W Va.
Youngstown Sheet & Tube Co., Youngstown, O.

BAR & ROD, STEEL, STAINLESS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Armco Steel Corp., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Henry Disston & Sons, Inc., Tacony, Philadelphia 35, Pa.
Firth-Sterling Steel & Carbide Corp., Demmler Rd., McKeesport, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.

BARE IRON PIPE COILS (See PIPE COILS, BENDS & BENDING)

BARE TUBE COILS

(A—Ammonia B—Other refrigerants)

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
American Brass Co., Waterbury 20, Ct. (*p. 189*)
(A,B) Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
(A,B) Bush Mfg. Co., 179 South St., W. Harftord 10, Ct. (*p. 40*)
(A,B) Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 81*)
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B) Frick Co., Waynesboro, Pa. (*p. 51*)
(A,B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(A,B) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(A,B) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 198*)
(B) Kramer Trenton Co., Olden & Brenning Aves., Trenton 5, N.J. (*p. 195*)
(B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. (*p. 41*)
(B) McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. (*p. 55*)
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa
(A,B) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(B) Reese & Long Refrigeration Products, Inc., 1891 Park Ave., N.Y.C. 35
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. (*p. 201*)
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
(A,B) Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
(A,B) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
(A) Roessing Mfg. Co., Sharpsburg St., Pittsburgh, Pa.
(B) Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Vilter Mfg Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 83*)
(A,B) Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
(A) Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. (*p. 52*)

BAROMETERS

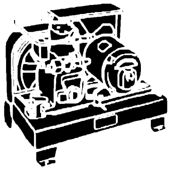
Bristol Co., Waterbury 20, Ct. (Recording only)
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Fee & Stemwedel, Inc., 2210 Wabansia Ave., Chicago 47, Ill.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Standard Thermometer, Inc., 952 Dorchester Ave., Boston 25, Mass.
Tagliabue Instruments Div. Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.

BASES, CABINET, COMPRESSOR, MOTOR, etc.

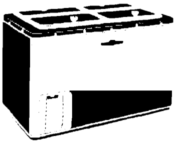
Ackermann Mfg. Co., Wheeling, W.Va.
Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. (*p. 178*)
American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Bossert Co., P.O. Drawer 358, Utica 1, N.Y.
Brasco Mfg. Co., Harvey, Ill.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7 Ill.
Evans Mfg. Co., 460 S. 10th Ave., Mt. Vernon, N.Y.
Fahlstrom Co., 13 Fahlstrom Ct., Passaic, N.J.
Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Rockwood Mfg. Co., 1801 English Ave., Indianapolis, Ind.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Southern Belting Co., 236 Forsythe St., S.W., Atlanta 2, Ga.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

Kelvinator Dependability...

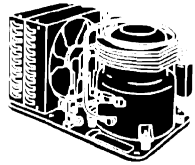
Proved by Time!



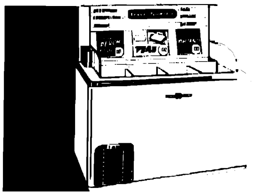
OPEN TYPE CONDENSING UNITS



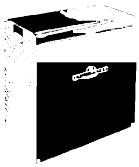
ICE CREAM CABINETS



SEALED TYPE CONDENSING UNITS

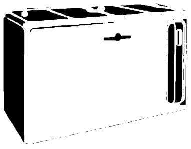


12 CU. FT. FROZEN FOOD MERCHANDISER



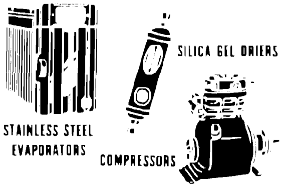
AIR DRIER

6 CU. FT. FROZEN FOOD MERCHANDISER



BEVERAGE COOLER

WATER COOLERS



GENUINE KELVINATOR PARTS AND SUPPLIES

YES! DEPENDABILITY is the "buy-word" of refrigeration customers today—you can give them DEPENDABILITY-PLUS with Kelvinator! Long life is literally engineered and built right into Kelvinator products. And if, after years of service, repair parts are needed—count on any one of Kelvinator's nation-wide network of Parts Depots to supply them—promptly and at reasonable prices. For complete information and prices on any of the products shown here—or for refrigeration supplies of any nature—call your nearest Kelvinator distributor. You'll find him in the classified section of your telephone directory.

PROFIT TODAY...BUILD FOR TOMORROW WITH

Kelvinator

THE NAME THAT SELLS...THE NAME THAT SATISFIES!

DIVISION OF NASH-KELVINATOR CORPORATION, DETROIT 32, MICHIGAN

SPECIALISTS IN PRECISION BUILT REFRIGERATION EQUIPMENT SINCE 1914

BASES, MACHINERY (See also VIBRATION ABSORBING BASES)

Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City 7, N.J.
Lukenweld, Inc., Coatesville, Pa.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.

BASKETS, WIRE (See WIRE FORMING)

BATCH FREEZERS (See FREEZERS)

BAUDELOT COOLERS (See WATER COOLERS, BAUDELOT TYPE)

BEARING MATERIAL

Bearium Metals Corp., 268 State St., Rochester, N.Y.
Belmont Smelting & Refining Works, Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
Eagle Picher Sales Co., American Bldg., Cincinnati 1, O.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13 (Weld Metal)
Federated Metals Div., American Smelting & Refining Co., 120 Broadway, N.Y.C. 5
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
International Powder Metallurgy Co., 439 W. Main St., Ridgway, Pa.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Mueller Brass Co., Port Huron, Mich.
National Lead Co., 111 Broadway, N.Y.C. 6
Northwest Lead Co., 2700-16th Ave., S.W., Seattle 4, Wash.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

BEARINGS, BALL

Ahlberg Bearing Co., 3025 W. 47th St., Chicago 32, Ill.
Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Bearings Co. of America, 501 Harrisburg Ave., Lancaster, Pa.
Boston Gear Works, Quincy 71, Mass.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Fafnir Bearing Co., 37 Booth St., New Britain, Ct.
Gwilliam Co., 360 Furman St., Brooklyn, N.Y.
Kaydon Engrg. Corp., Muskegon, Mich.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
McGill Mfg. Co., Inc., Valparaiso, Ind.
Marlin-Rockwell Corp., 402 Chandler St., Jamestown, N.Y.
Messinger Bearings, Inc., D St., above Erie Ave., Phila. 1, Pa.
Miniature Precision Bearings, Inc., Carpenter St., Keene, N.H.
New Departure Div., Gen'l. Motors Corp., Bristol, Ct.
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
Norma-Hoffman Bearings Corp., Hamilton Ave., Stamford, Ct.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa
Standard-Keil Hardware Mfg. Co., Inc., 639 Broadway, N.Y.C. 12
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

BEARINGS, BALL, FOR DRAWERS (See SLIDES)

BEARINGS, GRAPHITE, OILLESS OR SELF-LUBRICATING, etc.

Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Boston Gear Works, Quincy 71, Mass.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Bushings, Inc., 4358 Coolidge Highway, Royal Oak, Mich.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O.
Congress Drive Div., Tann Corp., 3750 E. Outer Drive, Detroit 34, Mich.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y
International Powder Metallurgy Co., 439 W. Main St., Ridgway, Pa.
Keystone Carbon Co., Inc., 1935 State St., St. Mary's, Pa.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Pure Carbon Co., Inc., 445 Hall Ave., St. Mary's, Pa.
Randall Graphite Bearings, Inc., Greenlawn & Erie RR., Lima, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
U. S. Graphite Co., Saginaw, Mich.
Wel-Met Co., 110 Gougler St., Kent, O.

BEARINGS, ROLLER

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
Charles Bond Co., 617 Arch St., Phila., Pa.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Fafnir Bearing Co., 37 Booth St., New Britain, Ct.
Gwilliam Co., 360 Furman St., Brooklyn, N.Y.
Hyatt Bearings Div., Gen'l. Motors Corp., Harrison, N.J.
Kaydon Engrg. Corp., Muskegon, Mich.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
McGill Mfg. Co., Inc., Valparaiso, Ind.
Marlin Rockwell Corp., Jamestown, N.Y.
Messinger Bearings, Inc., D St., above Erie Ave., Phila 1, Pa.
Norma-Hoffman Bearings Corp., Hamilton Ave., Stamford, Ct.
Roller Bearing Co. of America, Trenton, N.J.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa
V. E. Sprouse Co., Inc., 1804 22nd St., Columbus, Ind.

BEARINGS, SLEEVE

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
Ampco Metal, Inc., 1745 S. 38th St., Milwaukee 4, Wis.
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Chas. Bond Co., 617 Arch St., Phila., Pa.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Bushings, Inc. 4358 Collidge Highway, Royal Oak, Mich.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
International Powder Metallurgy Co., 439 W. Main St., Ridgway, Pa.
Kaydon Engrg. Corp., Muskegon, Mich.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.

(Continued)

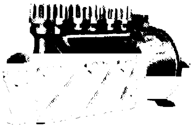
Westinghouse Electric Corporation

AIR CONDITIONING DIVISION

HYDE PARK *Offices and Distributors in All Principal Cities* BOSTON 36, MASS.

A complete line of Air Conditioning, Refrigeration, Heating, Ventilating and Electronic Air Cleaning Equipment including . . .

V and W Types
2, 3, 5, & 7½ h.p.



Inline Types
10, 15, 20, 25, 10 & 60 h.p.

V Type
75, 100 h.p.

HERMETICALLY-SEALED COMPRESSORS, TYPE CLS with direct-connected refrigerant-cooled motors. Refrigeration capacities from 2 to 100 tons (ASRE conditions) at suction temperatures from 10F to 50F. Ask for folder DB 101-110 and the CLS series.

WATER-COOLED CONDENSERS, TYPE CWC 13 sizes to economically match refrigeration requirements. 2 to 100 ton capacities.

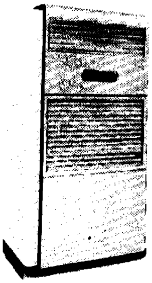
EVAPORATIVE CONDENSERS, TYPE EVA for applications where water for condensing purposes is limited. Up to 100 ton capacities. Ask for folder EVA-000-A.

WATER COOLERS, TYPE LC—vertical semi-flooded type for use in indirect systems. Capacities 7½ to 100 tons at a high rate of heat transfer. Ask for folder DB 104-105.

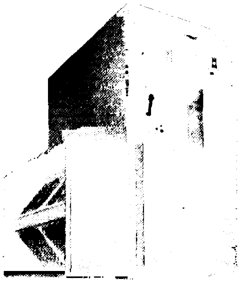
COOLING AND HEATING COILS—a complete line of Direct Expansion, Water and Steam Coils to meet all requirements. Coils designed for maximum heat transfer with staggered tubes and plate-type fins. Ask for 105-000 series.

AIR HANDLING UNITS, CENTRAL PLANT TYPE, with fan assembly of the backwardly inclined blade type, designed to accommodate cooling and heating coils, filters and humidifying equipment with capacities 1200 to 18,000 CFM. Ask for folders DB 91-750 and 102-720.

FOR SELF-CONTAINED INSTALLATIONS . . .



Self-Contained Unitaire ®



Central Plant Unitaire

SELF-CONTAINED UNITS, TYPE SU—capacities of 2, 3, 5 and 8 tons. Designed for within-the-space installations. May be remotely located for use with ductwork. Modern design cabinet blends with most interiors. Ask for folder DB 102-205.

CENTRAL PLANT UNITS, TYPE LU for remote installations. Capacities 10, 15, 20 and 25 tons. Ask for folder LU-000-A7.

Both the type SU and LU units are completely assembled, charged and tested at the factory and are ready for operation after simple connections to water, drain, power and ductwork (if used) are made.

To meet specific job requirements a full line of equipment is available, including Air Washers, Surface Dehumidifiers, Ventilating Fans and Electronic Air Cleaners (PRECIPITRON), providing a complete line of equipment from one source with undivided responsibility. Consult your Classified Telephone Directory for the Westinghouse Air Conditioning Distributor nearest you.

BEARINGS, SLEEVE (Continued)

McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
National Lead Co., 111 Broadway, N.Y.C. 6
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Randall Graphite Products Corp., 609 W. Lake St., Chicago 6, Ill.
Roller Bearing Co. of America, Trenton, N.J.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Saginaw Bearing Co., 820 S. Water St., Saginaw, Mich.
Shenango-Penn Mold Co., Dover, O.
V. E. Sprouse Co., Inc., 1804 22nd St., Columbus, Ind.

BEER COOLERS, BOTTLES (See BOTTLED BEVERAGE COOLERS)

BEER COOLERS, DRAFT

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Fleetwood Airflow, Inc., 421 N. Penna. Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 3, Tex.
(A,B,C) Gem Refrigerator Co., 165 W. Wyoming Ave., Phila. 40, Pa.
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
(B) John Herrell & Sons, Inc., 244 Lear St., Columbus 6, O.
(A,B) Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6, Mo.
(A) Kooler Keg Systems, 11 Mill St., Belleville 9, N.J.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
Lingle Refrigerator Co., Inc., 1116 E. 15th St., Kansas City 6, Mo.
(A,B) Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
National Cooler Corp., 1600 Woodland Ave., Cleveland, O.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nor-Lake, Inc., 2nd & Elm Sts., Hudson, Wis.
(A,B) R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Star Metal Mfg. Co., Inc., Trenton Ave., & Ann St., Phila. 34, Pa.
(B) Temprite Products Corp., E. Maple Road, Birmingham, Mich. *(p. 89)*
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 82)*
(A,B) United Refrigerator Co., Hudson, Wis.

BEER LINE INSULATION (See INSULATION)

BELLOWS, METALLIC

Bridgeport Thermostat Div. of Robertshaw-Fulton Controls Co., 1225 Connecticut Ave., Bridgeport 1, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Clifford Mfg. Co., 110 Grove St., Waltham 54, Mass
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Fulton Sylphon Co., Knoxville, Tenn.
Standard Thermometer, Inc., 952 Dorchester Ave., Boston 25, Mass.

BELLS (See SIGNALS)

BELTS & BELTING, FLAT

American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Baltimore Belting Co., 23 S. Gay St., Baltimore 2, Md.
Chas. Bond Co., 617 Arch St., Phila., Pa.
Boston Woven Hose & Rubber Co., P.O. Box 1071, Boston 3, Mass.
Cambridge Wire Cloth Co., Cambridge, Md.
Chicago Belting Co., 113 N. Green St., Chicago 7, Ill.
Chicago Rawhide Mfg. Co., 1306 Elston Ave., Chicago 22, Ill.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
R. & J. Dick Co., Inc., 24 Sade St., Passaic, N.J
Eagle Belting & Leather Co., 1401 Central Pkwy., Cin'ti. 14, O.
Fisher Leather Belting Co., Inc., 325 N. 3rd St., Phila. 6, Pa.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
L. H. Gilmer Co., Div. U. S. Rubber Co., 7230 Keystone St., Tacony, Phila. 35, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Graton & Knight Co., 356 Franklin St., Worcester 4, Mass.
Hewitt Rubber Div. Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Hudson Belting Co., 84 E. Worcester St., Worcester 4, Mass.
Manheim Mfg. & Belting Co., Manheim, Pa.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Quaker Rubber Corp., Tacony & Comly Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
J. E. Rhoads & Sons, 35 N. 6th St., Phila. 6, Pa.
Russell Mfg. Co., 400 E. Main St., Middletown, Ct.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Southern Belting Co., 236 Forsyth St., S.W., Atlanta 2, Ga.
Thermoid Co., Trenton, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N Y.C. 20

BELTS & BELTING, VEE

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Baltimore Belting Co., 23 S. Gay St., Baltimore 2, Md.
Boston Woven Hose & Rubber Co., P.O. Box 1071, Boston 3, Mass.
Browning Mfg. Co., Inc., Maysville, Ky.
Congress Drive Div., Tann Corp., 3750 E. Outer Drive Detroit 34, Mich.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Dayton Rubber Co., 2342 W. Riverview Ave., Dayton 1, O.
R. & J. Dick Co., Inc., 24 Sade St. Passaic, N.J.
Eagle Belting & Leather Co., 1401 Central Pkwy., Cin'ti. 14, O.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 171)*
L. H. Gilmer Co., Div. U. S. Rubber Co., 7230 Keystone St., Tacony, Phila. 35, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Maurey Mfg. Corp., 2907 S. Wabash Ave., Chicago 16, Ill.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Pyott Foundry & Machine Co., 328 N. Sangammon St., Chicago 7, Ill.
Quaker Rubber Corp., Tacony & Comly Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.

(Continued)

Carrier Corporation · Syracuse 1, New York

MARINE DIVISION
385 Madison Ave.
New York 17, N.Y.

Carrier

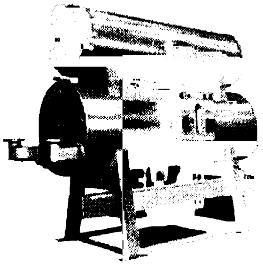
INTERNATIONAL DIVISION
385 Madison Ave.
New York 17, N.Y.

Offices and Dealers in all principal cities—refer to your telephone directory

Centrifugal Refrigerating Machines

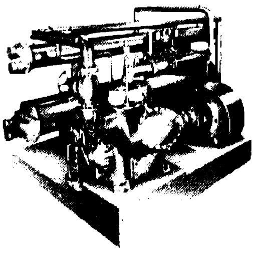
For large scale air conditioning, low temperature process refrigeration, direct or indirect cooling of water, brines, hydrocarbons and other liquids, and for liquefaction of vapors and gases. Capacities from 100 to 2000 tons cooling using steam turbine, electric motor or diesel engine drive.

Absorption Refrigerating Machines



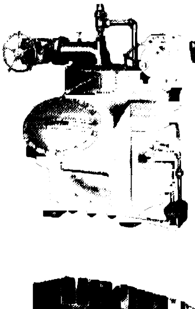
Complete packaged refrigerating plant for producing chilled water at 36F or higher in fully automatic operation between 10 and 100 percent capacity. Uses high or low pressure steam for operation with salt and water solution as the absorbent. Five sizes in cooling capacities from 115 to 350 tons.

Reciprocating Refrigerating Machines



Complete packaged refrigerating plant of 100 to 200 tons capacity. Chills water or brines in all ranges down to minus 20F and condenses vapors. Steam turbine, electric motor or diesel engine drive.

Compressors and Condensing Units



From 3 hp to 200 hp suitable for use with Freon-12, Freon-22, ammonia and Carrene-7 throughout entire temperature range of refrigeration and air conditioning applications. Direct or belt drive.

Blast Freezers and Cold Diffusers

For food freezing and storage, meat packing operations and other industries requiring medium and low temperatures. Units are available for brine, ammonia and Freon refrigerants, in suspension or floor models, dry coil and spray type in a range of sizes and capacities up to 40 tons.

Evaporative Condensers

For outdoor or indoor installation in air conditioning, refrigeration and industrial water cooling in place of water cooled condensers or cooling towers. Simplify water supply and disposal problems. Seven sizes in capacities from 10 to 90 tons.

Write for descriptive literature on any of the above equipment.

BELTS & BELTING, VEE (Continued)

Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Simonds Gear & Mfg. Co., 2501 Liberty Ave., Pittsburgh 22, Pa.
Thermoid Co., Trenton, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
V-Belt Clutch Co., 418 N. Western Ave., Los Angeles 4, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

BENDING (See PIPE COILS, BENDS & BENDING)

BENDING, SMALL PARTS (See STAMPINGS)

BENDS (See RETURN BENDS)

BERYLLIUM COPPER

Beryllium Corp., P.O. Box 1462, Reading, Pa.
Gen'l. Plate Div., Metals & Controls Corp., Attleboro, Mass.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

BEVERAGE COOLERS (See BOTTLED BEVERAGE COOLERS)

BEZELS (See NAMEPLATES)

BIMETALS (See THERMOSTATIC BIMETALS)

BLADES, FAN (See FAN BLADES)

BLAST COILS (See AIR CONDITIONING COILS)

BLAST FREEZERS (See FREEZERS)

BLENDERS, STEAM-WATER, etc.

Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*

BLOCK TIN (See TIN)

BLOOD PLASMA REFRIGERATORS (See REFRIGERATORS, BIOLOGICAL)

BLOWERS, BLOWER WHEELS, HOUSINGS & SCROLLS (See also FANS)

Advance Fan & Blower Co., 3428 Bagley, Detroit, Mich.
Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Allington & Curtis Mfg. Co., 1810 Holland Ave., Saginaw, Mich.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Machine Products Co., Marshalltown, Iowa
Bauer Bros. Co., 1774 Sheridan Ave., Springfield, O.
Bayley Blower Co., 1817 S. 66th St., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Campbell Heating Co., 3121 Dean St., Des Moines 4, Ia.
Carling Turbine Blower Co., 25 St. John's Rd., Worcester, Mass.
Century Fan Ventilator Co., 47 Cedar St., Stamford, Ct.
Champion Blower & Forge Co., Harrisburg & Charlotte Sts., Lancaster, Pa.
Chicago Blower Corp., 4588 W. Congress St., Chicago 24, Ill.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Commercial Shearing & Stamping Co., 1775 Logan St., Youngstown, O.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich. *(p. 29)*
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Fasco Industries, Inc., Union & Augusta Sts., Rochester 2, N.Y.
Garden City Fan Co., 332 S. Michigan Ave., Chicago 4, Ill.
General Blower Co., Ferris Ave., Morton Grove, Ill.
Hartzell Propeller Fan Co., Frances Ave., Piqua, O.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
King Co , 902 N. Cedar St., Owatonna, Minn.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Leiman Bros., Inc., 100 Christie St., Newark 5, N.J.
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa
Moore Co., Marceline, Mo.
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, O.
National Engrg. & Mfg. Co., 519 Wyandotte, Kansas City 6, Mo.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Newcomb-Detroit Co., 5741 Russell St., Detroit 11, Mich.
New York Blower Co., 3155 Shields Ave., Chicago 16, Ill.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Redmond Co., Inc., Owosso, Mich.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Robinson Ventilating Co., Zelienople, Pa.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
V. E. Sprouse Co., Inc., Columbus, Ind.
Standard Elec. Mfg. Co., W. Berlin, N.J.
Sterling Blower Co., 791 Windsor St., Hartford 1, Ct.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville, St., Boston 36, Mass. *(p. 23)*
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Torrington Mfg. Co., 70 Franklin St., Torrington, Ct. *(p. 27)*
Trane Co., La Crosse, Wis. *(p. 118)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
Utility Appliance Corp., 4851 S. Alameda St., Los Angeles 11, Cal.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

BLOWER FILTER UNITS

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
American Machine Products Co., Marshalltown, Ia.
Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Excelsior Steel Furnace Co., 118 S. Clinton, Chicago 6, Ill.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
King Co., 902 N. Cedar St., Owatonna, Minn.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
P. H. MaGirl Foundry & Furnace Wks., 413 E. Oakland Ave., Bloomington, Ill.
National Air Conditioning, Inc., Johnstown, Pa.
National Engrg. & Mfg. Co., 519 Wyandotte St., Kansas City 6, Mo.

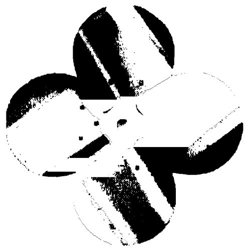
(Continued)

Vairified*
AIR IMPELLERS

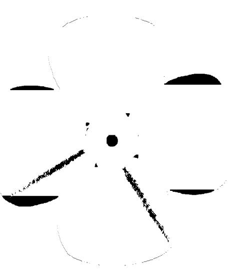
SHOULD BE SELECTED DURING EARLY STAGES OF UNIT DESIGN

By consulting Torrington during the early stages of product design, manufacturers planning the inclusion of an air impeller in a new unit will secure the greatest good from our long experience in the application of forced air.

The annoying, sometimes embarrassing delays and expense of design changes can be avoided with expert help at the right time, which, we repeat, is *in the beginning.*



From among the wide range of sizes, pitches and styles of Airistocrat propeller type fan blades, or sizes and widths of Airotor blower wheels, the correct air impeller for your needs can usually be supplied. Write or phone for information.



***Vairified** . . . aerodynamically designed for maximum operating efficiency with quietness; guaranteed to perform as rated under NAFM and NEMA Test Code conditions; hand gauged for uniform contour, alignment and balance. The Torrington Manufacturing Company.

TORRINGTON

MANUFACTURING COMPANY, TORRINGTON, CONN.

BLOWER FILTER UNITS (Continued)

Pace Co., 116 N.W. Couch, Portland, Ore.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
Viking Air Conditioning Corp,. 5600 Walworth Ave., Cleveland 2, O.

BODIES, TRUCK (See TRUCK BODIES, REFRIGERATED)

BOLTS, NUTS & SCREWS (See also FASTENERS)

Allen Mfg Co., 135 Sheldon St., Hartford 5, Ct.
Allmetal Screw Products Co., Inc., 33 Greene St., N.Y.C 13
American Screw Co., Willimantic, Conn.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11.
Bethlehem Steel Co., Bethlehem, Pa.
Boots Aircraft Nut Corp., 50 St. John St., Stanford, Ct.
Boss Nut & Bolt Co., 3403 W. 47th St., Chicago, Ill.
Bristol Co., Waterburg 20, Ct.
Buffalo Bolt Co., 101 East Ave., N. Tonawanda, N.Y.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Screw Co., 2701 Washington Blvd., Bellwood, Ill.
Clark Bros. Bolt Co., Milldale, Ct.
Cleveland Cap Screw Co., 2917 E. 79th St., Cleveland 4 O.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
Continental Screw Co., 459 Mt. Pleasant St., New Bedford, Mass.
Corbin Screw Corp, New Britain, Ct.
Elastic Stop Nut Corp., of America, 2330, Vauxhall Rd., Union, N.J.
Grip Nut Co., 310-RD S. Michigan Ave., Chicago, Ill.
H. M. Harper Co., 8200 Lehigh Ave., Morton Grove, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Holo-Krome Screw Corp., 25 Brook St., Hartford 10, Ct.
Laminated Shim Co., Inc., Union St., Glenbrook, Ct.
Lamson & Sessions Co., 1071 W. 85th St., Cleveland 2, O.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Oliver Iron & Steel Corp., S. 10th & Muriel Sts., Pittsburgh 3, Pa.
Wm. H. Ottemuller Co., Pattison St. & M.&P. R.R., York, Pa.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Parker-Kalon Corp., 200 Varick St., N.Y.C. 14
Penn Engrg. & Mfg. Corp., 305 Main St., Doylestown, Pa.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Prestole Corp., 1345 Miami St., Toledo, O.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Rockford Screw Products Co., 2501-9th St., Rockford, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Jos. T. Ryerson & Son, Inc., 16th & Rockwell St., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Shakeproof, Inc., 2401 N. Keeler Ave , Chicago 39, Ill.
Standard Pressed Steel Co., Box 824, Jenkintown, Pa.
Star Expansion Bolt Co., 147 Cedar St., N.Y.C. 6
Stover Lock Nut & Machinery Corp., Hilton & Holly Sts., Easton, Pa.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Thompson-Bremert Co., 1640 W. Hubbard St., Chicago 22, Ill.
Tinnerman Products, Inc., 8700 Brook Park Rd., Cleveland 13, O.
Townsend Co., 205 River Rd., New Brighton, Pa.
Triplex Screw Co., 5317 Grant Ave., Cleveland 4, O.
United-Carr Fastener Corp., 31 Ames St., Cambridge 42, Mass.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
J. H. Williams & Co., 400 Vulcan St., Buffalo 7, N.Y.

BOTTLED BEVERAGE COOLERS
(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) American Refrigeration Corp., 1025 E. Excelsior Ave., Hopkins, Minn.
(A) Artkraft Mfg. Corp., Kibby St. & D.T.&I. R.R., Lima. O.
(A) Bally Case & Cooler Co., Bally, Pa.
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(A,B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y.
(A) Crosley Div., Avco Corp., Cin'ti. 25, O.
(A,B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B,C) Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
(A,B) Esco Cabinet Co., West Chester, Pa.
(A,B) Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(C) Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis Md. *(p. 207)*
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 3, Tex.
(A,B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gem Refrigerator Co., 165 W. Wyoming Ave., Phila. 40, Pa.
(A) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
(A,B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(C) Ice Cooling Appliance Corp., Morrison, Ill.
(A,B) Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6, Mo
(A) Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(A) Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
(A,B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
(A,B) Lingle Refrigerator Co., Inc., 1116 E. 15th St., Kansas City 6, Mo.
(A,B) Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
Minneapolis Show Case & Fixture Co., New Richmond, Wis.
(A,B) Morton Show Cases, Inc., Washington Courthouse, O.
(A,B) John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
Nor-Lake, Inc., 2nd & Elm Sts., Hudson, Wis.
(A,B) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
(A,B) R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
(A) Portable Elevator Mfg. Co., 920 E. Grove St., Bloomington, Ill.
(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(A) S & S Products, Inc., Lima, O.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14. O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
(B) Southern Fixture Mfg. Co., P.O. Box 245 Greensboro, N.C.

(Continued)

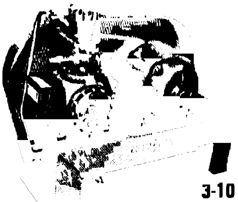
REFRIGERATION *and* AIR CONDITIONING

These products, developed by G. E.'s Climate Engineers, have built outstanding records for low-cost operation, dependability, and performance.

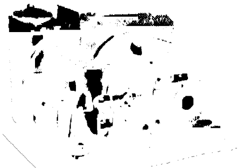
10-75 hp open type

G-E REFRIGERATION MACHINES

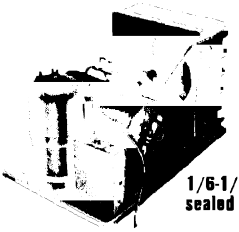
General Electric Condensing Units and Compressor Units are available in a wide range of models and sizes to fit the requirements of many different air conditioning and refrigeration applications, both commercial and industrial. Outstanding G-E advantages include: valve-in-head compression system; close-fitting pistons; wide-opening, tight-sealing valves; counter balanced crankshaft; G-E patented valve retainer.



3-10 hp open type



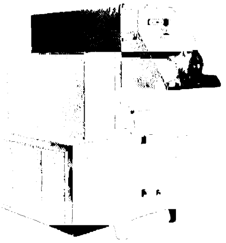
1/4-2 hp open type



1/6-1/2 hp sealed type

G-E AIR CONDITIONING

Central Plant Air Conditioners for large space applications — 5 basic frame sizes with capacity range from 1,000 to 12,800 CFM. G-E Personal Weather Control Systems for multi-room buildings, using individual room units from ⅝ to 1½ tons cooling capacity. G-E Packaged Air Conditioners in five sizes—2, 3, 5, 7½ and 10 hp.



G-E WATER COOLERS

Increase employee efficiency, improve public relations in offices, factories, and retail stores.

- **Air-cooled pressure models—4, 7, 10 gph**
- **Water-cooled pressure model—10 gph**
- **Air-cooled bottle model—3 gph**

GENERAL ELECTRIC COMPANY • AIR CONDITIONING DEPARTMENT • BLOOMFIELD, NEW JERSEY

For further information on these products, contact your nearest G-E Regional Office:

NEW YORK 22, N. Y., 570 Lexington Avenue
CHICAGO 54, ILL., 1127 Merchandise Mart
NEW ORLEANS 12, LA., 511-513 International Trade Mart
SAN FRANCISCO 6, CAL., 235 Montgomery Street

You can put your confidence in—

GENERAL ELECTRIC

BOTTLED BEVERAGE COOLERS (Continued)

(A,B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. (*p. 82*)
(A,B) United Refrigerator Co., Hudson, Wis.
(A,B) Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
(A,B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B,C) Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.
(A,B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(A,B) Wilson Cabinet Co., Inc., Smyrna, Del.

BRACKETS, SHELF (See HARDWARE SPECIALTIES)

BRASS (See particular mill form, i.e., BAR, SHEET, etc.)

BRAZING

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. (*p. 178*)
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.
Heintz Mfg. Co., Front St. & Olney Ave., Philadelphia 20, Pa.
Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 83*)
Queen City Steel Treating Co., 2980 Spring Grove Ave., Cin'ti. 25, O. (Elec. Furnace)
United Wire & Supply Corp., 1497 Elmwood Ave., Providence, 7 R.I.

BRAZING RODS (See WELDING ELECTRODES, RODS & WIRE)

BREAD COOLERS, BAKERY

Evans Mfg. Co., 460 S. 10th Ave., Mt. Vernon, N.Y.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. (*p. 207*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O.
Read Machinery Div., Standard Stoker Co., York, Pa.
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Tenney Engrg., Inc., 26 Ave. B., Newark 5, N.J.
Union Steel Products Co., 448 Pine St., Albion, Mich. (*p. 175*)
Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.

BREAKER STRIPS

Anchor Plastics Co., Inc., 533-541 Canal St., New York 13, N.Y.
Brasco Mfg. Co., Harvev, Ill.
Continental Diamond Fibre Co., 3 Chapel St., Newark, Del
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind.
Micarta Fabricators, Inc., 5324 N. Ravenswood Ave., Chicago 40, Ill.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Rigidized Metals Corp., 658 Ohio St., Buffalo 3, N.Y.
Sandee Mfg. Co., 5050 W. Foster Ave., Chicago 30, Ill.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Swan Rubber Co., Mansfield & Iron Aves., Bucyrus, O.
Amos Thompson Corp., Edinburg, Ind.
Tylac Co., Greeley & High St., Monticello, Ill.

BRINE CIRCULATORS (See AGITATORS)

BRINE COOLERS, DOUBLE TUBE

Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*),
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32) N.Y. (*p. 31*
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Reco Products Div., Refrigeration Engrg Corp., 2020 Naudain St., Phila. 46, Pa.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., 81 South St. Hartford 10, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO_2)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill
York Corp., York, Pa. (*p. 52*)

BRINE COOLERS, SHELL & COIL

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 23, N.Y. (*p. 31*)
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. (*p. 90*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fulleton, Cal. (*p. 39*)
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City 7, N.J
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. (*p. 89*)
Trane Co., La Crosse, Wis. (*p. 118*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO_2)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

BRINE COOLERS, SHELL & TUBE

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)

(Continued)

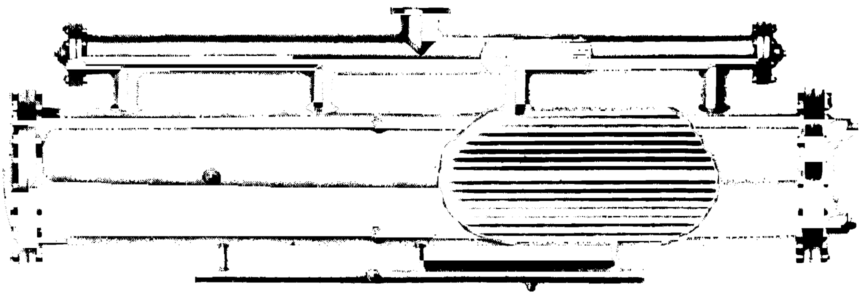
DOYLE & ROTH MFG. CO.

Foot of 23rd Street

BROOKLYN 32, N.Y.

SHELL & TUBE

HEAT TRANSFER EQUIPMENT



FLOODED WATER COOLER

Freon & Ammonia Condensers

Direct Expansion Water & Brine Coolers

Flooded Water & Brine Coolers

Chemical Process Coolers

SPECIAL COOLERS & CONDENSERS

D & R specializes in coolers & condensers
for applications not encountered ordinarily
in everyday refrigerating practice.

Write today for complete information on D. & R. products and services.

BRINE COOLERS, SHELL & TUBE (Continued)

Creamery Package Mfg. Co., 1234 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 81*)
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. (*p. 90*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nooter Corp., 1426 S. 2nd St., St. Louis 4. Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Reynolds Mfg. Co., Inc., Springfield, Mo.
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Ross Heater & Mfg. Co., Div. of Am'n. Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Trane Co., La Crosse Wis. (*p. 118*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 38*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

BRINE SPRAY UNITS (See UNIT COOLERS)

BRINE TANKS (See TANKS)

BRINE TREATMENT (See WATER TREATING)

BRONZE (See particular mill form, i.e., BAR, SHEET, etc.)

BRONZE, BEARING STOCK (See BEARING MATERIALS)

BRONZE PARTS (See particular part, also mill forms)

BRUSHES, MOTOR & GENERATOR

Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
North American Fibre Products Co., Standard Bldg., Cleveland 13, O.
Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Pure Carbon Co., Inc., 445 Hall Ave., St. Marys, Pa.
Speer Carbon Co., St. Marys, Pa.
Superior Carbon Products Co., 9115 George Ave., Cleveland 5, O.
Westinghouse Elec. Corp., 4454 Genessee St., Buffalo 5, N.Y.

BULK ICE MAKERS (See ICE MAKERS, BULK TYPE)

BUMPERS, RUBBER (See RUBBER PRODUCTS, MOLDED)

BUS AIR CONDITIONING SYSTEMS

Kero Industries, Inc., 2519 Gilbert Ave., Cincinnati 6, O.
Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.
York Corp., York, Pa.

BUSHINGS (See particular type; also BEARINGS, SLEEVE; also FITTINGS, PIPE)

BUSHINGS, PIPE (See FITTINGS)

BUSHINGS, RUBBER OR RUBBER BONDED (See RUBBER PRODUCTS, MECHANICAL BONDED)

BUTANE

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp. 30 E. 42nd St., N.Y.C. 17
Standard Oil of California, 225 Bush St., San Francisco 20, Cal.

CABINETS (See listings following, also particular type)

CABINETS, FOR AIR CONDITIONERS

Berger Mfg. Div., Republic Steel Corp., 1038 Belden Ave., N.E., Canton, O.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y
Erie Art Metal Co., 1602 E. 18 St., Erie, Pa.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20 Pa.
Hugo Mfg. Co., 49 Ave., W., & Superior St., Duluth 7, Minn.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Pare Co., 116 N.W. Couch, Portland, Ore.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Amos Thompson Corp., Edinburg, Ind.

CABINETS, FOR HOUSEHOLD REFRIGERATORS

American Central Mfg. Corp., Div. of Avco Corp., 18th & Columbia Sts., Connersville, Ind.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis Md. (*p. 207*)
Hugo Mfg. Co., 49 Ave., W., & Superior St., Duluth 7, Minn.
Rex Mfg. Co., Inc., Western Ave., Connersville, Ind.
Sanitary Refrigerator Co., Fond du Lac, Wis.
Seeger Refrigerztor Co., 850 Arcade St., St. Paul 6, Minn.
Stoddard Mfg. Co., 617-4th St., S.W., Mason City, Ia.
Wilson Refrigeration, Inc., S. Main St., Smyrna, Del.

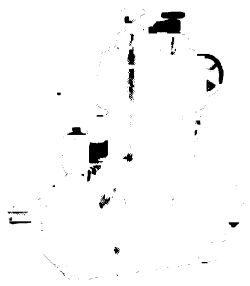
CABINETS, FREEZING (See HOME & FARM FREEZERS)

CABINET TOPS & LIDS

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40 Ill.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. (*p. 152*)
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24 Ill.
Formica Co., 4613 Spring Grove Ave., Cincinnati 32, O
Franklin Body & Equipment Corp., 1042 Dean St. Brooklyn 16, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jordon Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, Pa.
Luzerne Rubber Co., Trenton 9, N.J.
Panelyte Div., St. Regis Paper Co., 230 Park Ave. N.Y.C. 17
Stokes Molded Products, Inc., Taylor at Webster St. Trenton 4, N.J.
Thermacote Co., 1005 Santa Fe St., Los Angeles 21, Cal.

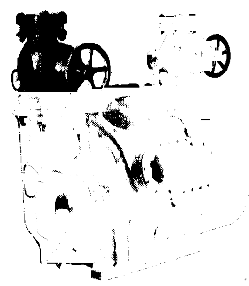
CABLE, ELECTRIC (See WIRE, ELECTRIC)

Whatever Your Needs in Refrigeration—
You Can DEPEND on VILTER



VERTICAL COMPRESSORS

Ammonia units in sizes from 4 to 100 tons each. Modern design freon units with rotary shaft seals, from ¼ h.p. to 50 h.p. Horizontal compressors from 50 tons upward, for electric or steam drive.



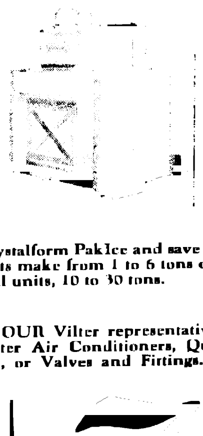
VMC COMPRESSORS

Latest type multi-cylinder compressors, with 4, 6, or 8 cylinders, for freon or ammonia. Completely balanced, VMC's require no expensive foundation. All Vilter units are built *entirely* by Vilter.

BOOSTER COMPRESSORS

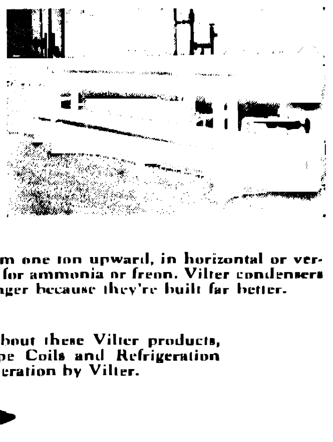
Champions of economy for low temperature refrigeration. Sizes range from 60 to 600 cubic feet per minute in vertical units. Larger capacities in two-stage horizontals. Completely Vilter-built as usual.

THE VILTER PAKICER



Make your own Crystalform PakIce and save money. Vertical PakIce units make from 1 to 6 tons of ice in 24 hours. Horizontal units, 10 to 30 tons.

SHELL AND TUBE CONDENSERS



Any size from one ton upward, in horizontal or vertical design, for ammonia or freon. Vilter condensers stand up longer because they're built far better.

SEE YOUR Vilter representative for more information about these Vilter products, or Vilter Air Conditioners, Quick Freezing Systems, Pipe Coils and Refrigeration Piping, or Valves and Fittings. You're ahead with refrigeration by Vilter.

WRITE FOR CATALOGS

. . . on any of these Vilter products. Longer life and less trouble save you money with Vilter.

Vilter

REFRIGERATION and AIR CONDITIONING

THE VILTER MANUFACTURING COMPANY
MILWAUKEE 7, WISCONSIN

Ammonia and Freon Compressors • Pak Icers • Evaporative and Shell and Tube Condensers • Pipe Coils • Valves and Fittings

CALCIUM CHLORIDE

Barada & Page, Inc., Guinotte & Michigan Aves., Kansas City 1, Mo.
Henry Bower Chemical Mfg. Co., Gray's Ferry Rd. & 29th St., Phila. 46, Pa.
Columbia Chemical Div., Pittsburgh Plate Glass Co., 5th Ave. at Bellefield, Pittsburgh 13, Pa.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Michigan Alkali Div., Wyandotte Chemicals Corp., Wyandotte, Mich.
Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Phila. 7, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Solvay Sales Div., Allied Chemical & Dye Corp., 40 Rector St., N.Y.C. 6
Tamms Industries, 228 N. La Salle St., Chicago 1, Ill.

CALCIUM SULFATE, ANHYDROUS (See DEHYDRANTS)

CAN FILLERS (See ICE MANUFACTURING ACCESSORIES)

CANDY CASES, REFRIGERATED

Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
Harder Refrigerator Corp., Tyler Fixture Corp. Coldeskill, N.Y. *(p. 82)*
C. F. Moores Co., Inc., 1133 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.

CANS, ICE (See ICE CANS)

CANVAS & DUCK & SPECIALTIES

Canvas Products Co., 1236 S. 7th St., St. Louis 4, Mo.
Flexible Tubing Corp., New Whitfield St., Guilford, Ct.
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Wagner Awning & Mfg. Co., 2658 Scranton Rd., Cleveland 1, O.
Webb Mfg. Co., 4th & Cambria Sts., Phila. 33, Pa.

CAPACITORS, ELECTRICAL

Aerovox Corp., New Bedford, Mass.
Cornell-Dubillier Elec. Corp., S. Plainfield, N.J.
Fansteel Metallurgical Corp., North Chicago, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
P. R. Mallory & Co., Inc., 3029 E. Washington St. Indpls., Ind.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Sprague Products Co., Marshall St., N. Adams, Mass.
Tobe Deutschman Corp., 921 Providence Pike, Norwood, Mass.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

CAPS, WELDING (See FITTINGS, WELDING)

CARBON DIOXIDE (See also DRY ICE)

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Mathieson Chemical Corp., Mathieson Bldg., Baltimore 3, Md.
Pure Carbonic, Inc., 60 E. 42nd St., New York, 17, N.Y.

CARBON PRODUCTS (See also GRAPHITE PRODUCTS; also BRUSHES)

Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Pure Carbon Co., Inc., 445 Hall Ave., St. Marys, Pa.
Speer Carbon Co., St. Marys, Pa.

CARBONATORS & CARBONATING EQUIPMENT

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Economy Faucet Co., 12 New York Ave., Newark, N.J.
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
Hudson Industries, Inc., 4400 St. Aubin Ave., Detroit 7, Mich.
Liquid Carbonic Corp., 3100 S. Kezdie Ave., Chicago 23, Ill.
Mojonnier Bros. Co., 4601 W. Ohio St., Chicago, Ill.
Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 45, Wis.
Scientific Research Co., 1618 N. Vancouver, Portland 12, Ore.
Spacarb, Inc., 311 E. 23rd St., N.Y.C. 10
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p. 89)*
Wooster Brass Co., 1415 E. Bowman St., Wooster, O

CARTS, MATERIAL HANDLING (See TRUCKS)

CASTINGS, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
James H. Matthews & Co., 3948 Forbes St., Pittsburgh 13, Pa.
Newman Bros., Inc., 666 W. 4th St., Cincinnati 3, O.
Paxton-Mitchell Co., 2614 Martha St., Omaha 5, Neb.
Permold Co., W. Liberty St., Medina, O.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.
Wooster Brass Co., 1415 E. Bowman St., Wooster, O

CASTINGS, BRASS, BRONZE, etc.

Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bethlehem Steel Co., Bethlehem, Pa.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Hills-McCanna Co., 3029 N. Western Ave., Chicago 18, Ill.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Manistee Iron Wks. Co., River St., Manistee, Mich.
James H. Matthews & Co., 3948 Forbes St., Pittsburgh 13, Pa.
Mueller Brass Co., Port Huron, Mich.
National Lead Co., 111 Broadway, N.Y.C. 6
Newman Bros., Inc., 666 W. 4th St., Cin'ti. 3, O.
Niles Tool Wks. Div., General Machinery Corp., 545 N 3rd St., Hamilton, O.
Paxton-Mitchell Co., 2614 Martha St., Omaha 5, Neb.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.
York Corp., York, Pa. *(p. 52)*
J. A. Zurn Mfg. Co., Erie, Pa

CASTINGS, DIE

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2, Mich.
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.

(Continued)

What every refrigeration man should know about "VIRGINIA" products

"VIRGINIA'S" NEWEST PRODUCT CAN-O-GAS

Handy, throw-away, no-deposit can the perfect way to charge hermetic systems, water or beverage coolers, vendors. Available filled with "Freon-12" (15 oz.) or "Freon-114" (16 oz.).

"EXTRA DRY ESOTOO" (B.P. +14°F.)

"Extra Dry" is the refrigeration grade SO_2 that service and maintenance engineers have endorsed for more than 20 years. Comes in all popular cylinder sizes.

"V-METH-L" (B.P.–10.7°F.)

Virginia Methyl Chloride is made specifically for refrigeration use. Low moisture content, low acidity and narrow boiling range recommend "V-Meth-L" for the most exacting requirements.

"FREON" REFRIGERANTS

"FREON-11" "Boiling Point" 74.7° F.	"FREON-12" "Boiling Point" 21.6° F.
"FREON-22" "Boiling Point" 41.4° F.	"FREON 113" "Boiling Point" 117.6° F.

"FREON-114"
"Boiling Point"
38.0° F.

Virginia Smelting Company is distributor for Kinetic's "Freon" Refrigerants.

PRESSTITE PRODUCTS

PERMAGUM. The perfect sealing compound for cases, inspection plates, pipe openings. Adheres to any dry surface; odorless; stays plastic 0 F. to 350°F. Brown Permagum comes in 2½ lb. and 45 lb. slugs; gray-white Permagum in 80 ft. rolls of 3/16″ cord.

INSULATION TAPE. Stops dripping pipes, valves and fittings permanently. Convenient package contains 30 ft. of 2″ wide rolls ⅛″ thick. Contains 40 percent virgin cork; adheres to any surface; joints are self-sealing.

ENAMELITE. Asphaltic mastic which doesn't have to be heated. Sticks to any dry surface; makes an excellent vapor seal. Effective on corkboard, Styrofoam, Celotex, Fiberglas. Enables you to quickly recondition cabinets, cold rooms, ducts. Excellent for small repair jobs.

VIRGINIA SMELTING COMPANY
WEST NORFOLK, VIRGINIA
PHILADELPHIA • NEW YORK • BOSTON
CHICAGO • DETROIT • ATLANTA

VIRGINIA
VIRGINIA Refrigerants

CASTINGS (Continued)

Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
Doehler-Jarvis Corp., 386-4th Ave., N.Y.C. 16
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Electric Auto-Lite Co., Instrument & Gauge Div. Champlain St., Toledo 1, O.
Frick Co., Waynesboro, Pa. (*p. 51*)
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. (*p. 121*)
Sterling Die Casting Co., Inc., 743-30th St., Brooklyn 32, N.Y.
Titan Metal Mfg. Co., Bellefonte, Pa.
York Corp., York, Pa. (*p. 52*)

CASTINGS, LEAD

Belmont Smelting & Refining Wks., 330 Belmont Ave., Brooklyn 7, N.Y.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6

CASTINGS, MONEL; NICKEL & NICKEL ALLOY

Cooper Alloy Foundry Co., Bloy St. & Ramsey Ave., Hillside 5, N.J.
Driver-Harris Co., Harrison, N.J.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
Newman Bros., Inc., 666 W. 4th St., Cincinnati 3, O.
Ohio Steel Foundry Co., James St., Springfield, O.
Ross-Mehan Foundries, Chattanooga, Tenn.

CASTINGS, STEEL & IRON

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Bonney-Floyd Co., 611 Marion Rd., Columbus, O.
Cooper-Bessemer Corp., Mt. Vernon, O.
Dayton Malleable Iron Co., P.O. Box 980, Dayton 1, O.
Benjamin Eastwood Co., 300 Straight St., Paterson, N.J.
Eaton Mfg. Co., 9771 French Rd., Detroit 33, Mich.
Frick Co., Waynesboro, Pa. (*p. 51*)
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Lakeside Malleable Castings Co., Racine, Wis.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
P. H. MaGirl Foundry & Furnace Wks., 413 E. Oakland Ave., Bloomington, Ill.
National Malleable & Steel Castings Co., 10600 Quincy Ave., Cleveland 6, O.
Niles Tool Wks. Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Ohio Steel Foundry Co., James St., Springfield 1, O.
Paxton-Mitchell Co., 2614 Martha St., Omaha 5, Neb.
Pyott Foundry & Machine Co., 328 N. Sangamon St., Chicago 7, Ill.
Ross-Mehan Foundries, Chattanooga, Tenn.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
Shenango-Penn Mold Co., Dover, O.
Union Mfg. Co., New Britain, Ct.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
York Corp., York, Pa. (*p. 52*)
J. A. Zurn Mfg. Co., Erie, Pa.

CEMENT, INSULATION (See also INSULATION ERECTION MATERIALS)

Alton Mineral Wool Insulation Co., 2317 Whitemore Place, St. Louis 4, Mo.
American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Carney Co., Inc., 125 S. 5th St., Minneapolis 2, Minn.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
International Vermiculite Co., Girard, Ill.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. (*p. 128*)
Pecora Paint Co., 3501 N. Fourth St., Philadelphia 40, Pa.
Philip Carey Mfg. Co., Lockland, Cincinnati 15, O.
Refractory & Instulation Corp., 120 Wall St., N.Y.C. 5
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Stancal Asphalt and Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Taft-Jenkins Co., Inc., 27 Sargeant St., Holyoke, Mass.
Virginia Smelting Co., W. Norfolk, Va. (*p. 35*)
Grant Wilson, Inc., 315 S. Sherman, Chicago 4, Ill.
Zonolite Co., 135 S. La Salle St., Chicago 3, Ill.

CEMENTS, RUBBER, SYNTHETIC, etc. (See ADHESIVES; also COMPOUNDS)

CHAIN, CONVEYOR, DRIVE, ROLLER, etc.

Alvey-Ferguson Co., Disney & R Sts., Cin'ti. 9, O.
Baldwin Duckworth Div., Chain Belt Co., 369 Plainfield St., Springfield, Mass.
C. O. Bartlett & Snow Co., 6200 Harvard Ave., Cleveland 5, O.
Chain-Belt Co., 1600 W. Bruce St., Milwaukee 4, Wis.
Conveyor Co., 3260 E. Slauson Ave., Los Angeles 58, Cal.
Corbin Screw Corp., New Britain, Ct.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Diamond Chain Co., Inc., 402 Kentucky Ave., Indpls. 7, Ind.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.
Simonds Gear & Mfg. Co., 2501 Liberty Ave., Pittsburgh 22, Pa.
Transue & Williams Steel Forging Corp., Alliance, O.
United Engine Co., W. Holmes Rd., Lansing 12, Mich.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.

CHANNELS (See SHAPES, STRUCTURAL)

CHARGING LINES (See also FLEXIBLE CONNECTIONS; also TUBING, FLEXIBLE)

Aeroquip Corp., 300 S. East Ave., Jackson, Mich.
Allin Mfg. Co., 1153 W. Grand Ave., Chicago, Ill.
American Brass Co., Waterbury 88, Ct. (*p. 189*)
Chicago Metal Hose Corp., Maywood, Ill.
Fine Products Co., 185 N. Wabash Ave., Chicago I, Ill. (*p. 163*)
Flexible Tubing Corp., N. Main St., Branford, Ct.
Henry Valve Co., Melrose Park, Ill. (*p. 88*)
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Jarrow Products, 420 N. La Salle St., Chicago, Ill. (*p. 85*)
Johnson Metal Hose Co., 226 Mill St., Waterbury 88, Ct.
Madden Brass Products Co., 1111 N. Franklin St., Chicago 10, Ill.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. (*p. 88*)
Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.
York Corp., York, Pa. (*p. 52*)

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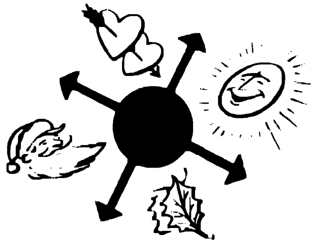

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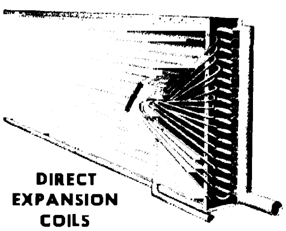
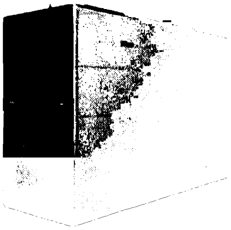
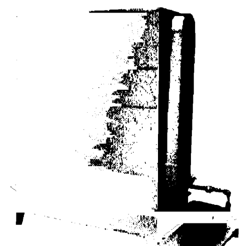


DIRECT EXPANSION COILS



REFRIGERATED KOOLER-AIRE



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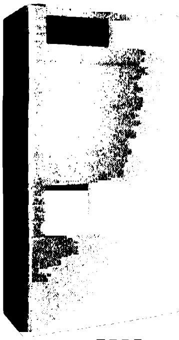
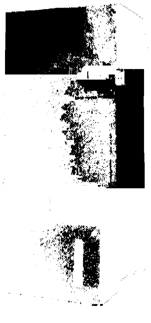


STORE CONDITIONER



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CHECK VALVES, AIR, WATER

Automatic Temperature Control Co., Inc., 5212 Pulaski Rd., Phila. 44, Pa.
Catawissa Valves & Fittings Co., Catawissa, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Conoflow Corp., 2100 Arch St., Phila. 3, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave. St. Louis 2, Mo.
R. & J. Dick Co., Inc., 24 Sade St., Passaic, N.J.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Grove Regulator Co., 6520 Hollis St., Oakland, Cal.
Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Ludlow Valve Mfg. Co., Foot of Adams St., Troy, N.Y.
Lunkenheimer Co., Beekman St., & Waverly Ave., Cin'ti. 14, O.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Swaby Mfg. Co., 3818 W. Armitage Ave., Chicago 47, Ill.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. (*p. 111*)
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 33, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
J. A. Zurn Mfg. Co., Erie, Pa.

CHECK VALVES, REFRIGERANT
(A—Ammonia; B—Other refrigerants)

(A,B) Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
(A,B) Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
(A,B) Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
(A,B) Frick Co., Waynesboro, Pa. (*p. 51*)
(A,B) Henry Valve Co., Melrose Park, Ill. (*p. 88*)
(A,B) Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. (*p. 189*)
(B) Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
(B) Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
(B) Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
(A,B) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
(B) Madden Brass Products Co., 1111 N. Franklin St., Chicago 10, Ill.
(B) Mueller Brass Co., 1025 Lapeer Ave., Port Huron, Mich.
(A,B) H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. (*p. 142*)
(B) Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. (*p. 88*)
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
(A) Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
(A,B) Watson-Stillman Co., Roselle, N.J. (*p. 111*)
(A,) Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. (*p. 52*)

CHILLERS, CONTINUOUS

Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Votator Div., Girdler Corp., Louisville 1, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

CIRCUIT BREAKERS

Allen-Bradley Co., Milwaukee 4, Wis.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Buchanan Elec'l Prod. Corp., 225 Route 29, Hillside, N.J.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Fasco Industries, Inc., Union & Augusta Sts., Rochester 2, N.Y.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
I.T.E. Circuit Breaker Co., 19th & Hamilton St., Phila., Pa.
Russell & Stoll Co., Inc., 125 Barclay St., N.Y.C.
Soreng Mfg. Corp., 9555 Eden Ave., Schiller Park, Ill.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Westinghouse Elec. Corp., Beaver, Pa.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

CIRCULATORS, AIR (See FANS)

CLAMPS, CONDUIT, CABLE, PIPE, TUBING, etc. (See also HANGERS, PIPE)

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Prestole Corp., 1345 Miami St., Toledo, O.
M. B. Skinner & Co., South Bend 23, Ind.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 13, O.
Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12, Wis.
U. S. Expansion Bolt Co., York, Pa.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

CLOCKS

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Elgin Nat'l. Watch Co., 103 National St., Elgin, Ill.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Marshalltown Mfg. Co., Marshalltown, Ia.

CLOSERS, COLD STORAGE DOOR

Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, O.
Chicago Spring Hinge Co., 1500 Carroll Ave., Chicago 7, Ill.
Jamison Cold Storage Door Co., Hagerstown, Md.
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.
Streator Products Corp., 508 N. 8th St., Fairfield, Ia.
York Corp., York, Pa. (*p. 52*)

CLOTH, WIRE (See MESH, METAL; also WIRE CLOTH)

CLUTCHES

Black-Clawson Co., Middletown, O.
Centric Clutch Co., Woodbridge, N.J.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Hilliard Corp., 102 W. 4th St., Elmira, N.Y.
W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago 24, Ill.

Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Mercury Clutch Div., Automatic Steel Products, Inc., 1201 Camden Ave., Canton 6, O.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Twin Disc Clutch Co., Racine, Wis.
V-Belt Clutch Co., 418 N. Western Ave., Los Angeles 4, Cal.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.

COATINGS & COMPOUNDS, MASTIC, PROTECTIVE, WATERPROOFING, etc.

American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
American Chemical Paint Co., Ambler, Pa.
American Pipe & Construction Co., Box 3428 Terminal Annex, Los Angeles 54, Cal.
Arco Co., 7301 Bessemer Ave., Cleveland 27, O.
Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Atlas Mineral Products Co., Mertztown, Pa.
Atlas Powder Co., N. Chicago, Ill.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
Dampney Co. of America, Hyde Park, Boston 36, Mass.
Dennis Chemical Co., 2701 Papin St., St. Louis 3, Mo.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Esso Standard Oil Co., 15 W. 51st St., N.Y.C. 19
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Glidden Co., 11001 Madison Ave., Cleveland 2, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
A. C. Horn Co., Inc., 10th Street & 44th Ave., Long Island City 1, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
Master Mechanics Co., Freeman Rd., Cleveland 13, O.
Minnesota Mining & Mfg. Co., 900 Fauquier St., St. Paul 6, Minn.
Miracle Adhesives Corp., 214 E. 53rd St., N.Y.C. 22
J. W. Mortell Co., Burch St., Kankakee, Ill.
North American Mogul Products Co., Standard Bldg., Cleveland 13, O.
NuKem Products Corp., 111 Colgate Ave., Buffalo 20, N.Y.
Oakite Products, Inc., 22 Thames St., N.Y.C. 6
Pecora Paint Company, 3501 N. Fourth St., Philadelphia 40, Pa.
Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Philadelphia 7, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Plaskon Div. Libbey Owens Ford Glass Co., 2112 Sylvan Ave., Toledo 6, O.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 *(p. 136)*
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
Stancal Asphalt & Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
Tempo Chemical Co., Inc., 4702 5th St., Long Island City 1, N.Y.
United Chromium, Inc., 100 E. 42nd St., N.Y.C. 17
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.
Virginia Smelting Co., W. Norfolk, Va. *(p. 85)*
Vita-Var Corp., 1186 Raymond Blvd., Newark 2, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
X-Pando Corp., 43-15-5[illegible]h St., Long Island City 1, N.Y

COCKS

Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 33, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

COIL & BAFFLE ASSEMBLIES
(A—Ammonia; B—Other refrigerants)

American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B) Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
(B) Betz Corp, 445 State St., Hammond, Ind.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
Cold Control, Inc., 111 Broadway, N.Y.C. 6
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
A,B) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
(B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 195)*
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(B) McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. *(p. 55*

(Continued)

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COIL & BAFFLE ASSEMBLIES (Continued)

(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(B) Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N.J.
(A,B) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(B) Reese & Long Refrigeration Products, Inc., 1891 Park Ave., N.Y.C. 35
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(A,B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
(A,B) Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

COIL STOCK (See STRIP)

COILS (See particular type, i.e., FINNED COILS; PLATE COILS; UNIT COOLERS; etc.)

COILS, SPECIAL
(A—Ammonia; B—Other refrigerants)

(A,B) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
(A,B) Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
(A,B) Baker Refrigeration Corp. Inc., S. Windham, Me. *(p. 63)*
(B) Betz Corp., 445 State St., Hammond, Ind.

(Continued)

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- ✓Refrigeration of Aluminum Alloys
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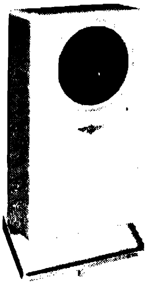
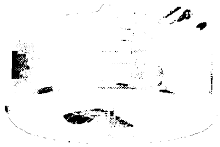
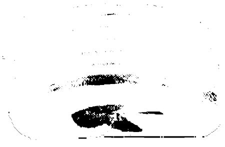


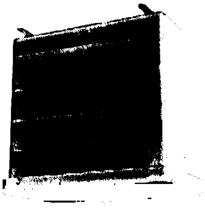
WALL HUMI-TEMP



TURRET HUMI-TEMP

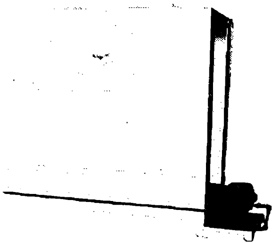


HALF-TURRET HUMI-TEMP



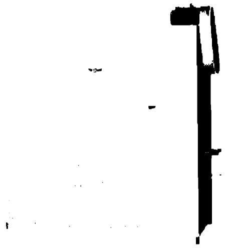
STANDARD HUMI-TEMP
ALSO
LOW TEMPERATURE UNITS
WITH AUTOMATIC HOT
GAS DEFROSTER

• For Air Conditioning Equipment



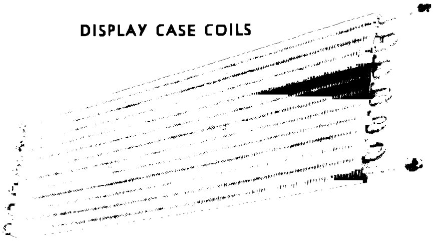
COOLING TOWERS & EVAPORATIVE CONDENSERS

AIR CONDITIONING UNITS

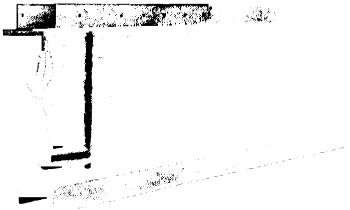


FLOOR HUMI-TEMP UNITS

• For Original, Cross-Fin Aluminum Coils



DISPLAY CASE COILS



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COILS, SPECIAL (Continued)

(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
(B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y.
(A,B) Crane Co., 836 Michgan Ave., Chicago 5, Ill. *(p. 109)*
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
(A) Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Dole Refrigerating Co., 5910 N. Puleski Rd., Chicago 30, Ill.
(A,B) Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B) Frick Co., Waynesboro, Pa. *(p. 51)*

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(B) Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 88)*
(A,B) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
(A,B) Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 158)*
(A,B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 195)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(B) Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
(B) McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. *(p. 55)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
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(A,B) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
(A) Nooter Corp., 1426 S. 2nd St., St.Louis 4, Mo.
(A) Pittsburgh Pipe Coil & Bending Co., 61 Bridge St. Etna P.O., Pittsburgh 23, Pa.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(B) Reese & Long Refrigeration Products, Inc., 1891 Park Ave., N.Y.C. 35
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(A,B) Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Edw. Renneburg & Sons So., 2639 Boston St., Baltimore 24, Md.
(A,B) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
(A) Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
(B) Rome-Turney Radiator Co., Rome, N.Y. *(p. 105)*
(A,B) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(A,B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 88)*
(A) Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
(A,B) Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 52)*
Young Radiator Co., Racine, Wis. *(p. 77)*

COLD CONTROLS (See THERMOSTATS)

COLD STORAGE DOORS & WINDOWS

Butcher Boy Cold Storage Door Co., 170 N. Sangamon St., Chicago 7, Ill. *(p. 45)*
Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17

(Continued)

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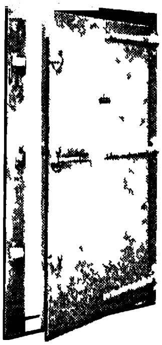
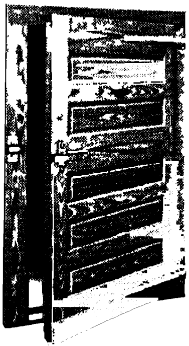
For Moderately Low Temperatures



SERIES "50" ***Monopanel Construction***



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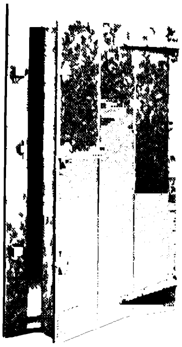
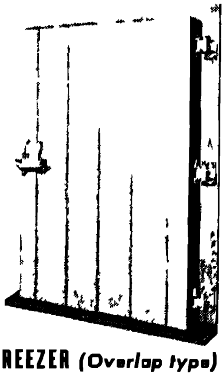
Raised Panel Construction

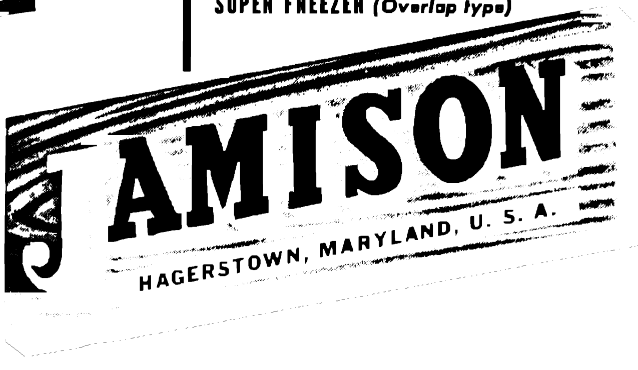
For Zero and Sub-Zero Use



LO-TEMP ***(Infitting type)***



SUPER FREEZER ***(Overlap type)***



COLD STORAGE DOORS (Continued)

Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Cruse Refrigerator Co., Inc., 504 Main St., Louisville 2, Ky.
Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. (*p. 207*)
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
Jamison Cold Storage Co., Hagerstown, Md. (*p. 43*)
Jordon Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, Pa.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Masterfreeze Corp., Sister Bay, Wis.
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 (*p. 136*)
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Streator Products Corp., 408 North Delpot St., Fairfield, Ia.
Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.
York Corp., York, Pa. (*p. 52*)

COLD STORAGE DOOR HARDWARE (See HARDWARE, COLD STORAGE DOOR; also CLOSERS)

COLLARS, SHAFT

Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Standard Pressed Steel Co., Box 824, Jenkintown, Pa.

COLUMNS, WATER (See GAUGE GLASSES)

COMPOUNDS, CAULKING

Calbar Paint & Varnish Co., 2612 N. Martha St., Phila. 25, Pa.
Jesse G. Croll & Son, 2606 Germantown Ave., Phila. 33, Pa.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
Lexington Supply Co., 4815 Lexington Ave., Cleveland 3, O.
Master Mechanics Co., Freeman Rd., Cleveland 13, O.
Midland Paint & Varnish Co., 3801 E. 91 St., Cleveland 5, O.
J. W. Mortell Co., Burch St., Kankakee, Ill.
Nebell Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Pecora Paint Co., Inc., 3501 N. Fourth St., Philadelphia 40, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
Virginia Smelting Co., W. Norfolk, Va. (*p. 35*)
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
X-Pando Corp., 43-15-36th St., Long Island City 1, N.Y.

COMPOUNDS, CORROSION INHIBITING (See WATER TREATING)

COMPOUNDS, GASKET, PIPE JOINT, etc.

Arco Co., 7301 Bessemer Ave., Cleveland 27, O.
Armstrong Cork Co., Lancaster, Pa.
Atlas Mineral Products Co., Mertztown, Pa.
Calbar Paint & Varnish Co., 2612 N. Martha St., Phila. 25, Pa.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Chicago Gasket Co., 1275 West North Ave., Chicago, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Crane Packing Co., 1800 Cuyler Ave., Chicago 13, Ill.
Jesse G. Croll & Son, 2606 Germantown Ave., Phila. 33, Pa.
Dearborn Chemical Co. Merchandise Mart Plaza, Chicago 54, Ill.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
Goetze Branch, Johns-Manville Products Corp., Allen Ave., New Brunswick, N.J.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Highside Chemicals Co., 10 Colfax Ave., Clifton, N.J.
A. C. Horn Co., Inc., 10th Street & 44th Ave., Long Island 1, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Keystone Lubricating Co., 21st & Lippincott St., Phila. 32, Pa.
Lake Chemical Co., 3052 W. Carroll Ave., Chicago 12, Ill.
Macksons Co., 127 Cedar St., N.Y.C. 6
Minnesota Mining & Mfg. Co., 900 Fauquier St., St. Paul 6, Minn.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Pecora Paint Co., Inc., 3501 N. Fourth St., Phila. 40, Pa.
Permatex Co., Inc., 1720 Ave. Y, Brooklyn 29, N.Y.
Presstite Engrg. Co., 2900 Chouteau Ave., St. Louis 10, Mo.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
H. H. Robertson Co., 2400 Farmers Bank Bldg., Pittsburgh 22, Pa.
Rutland Fire Clay Co., Curtis Ave., Rutland, Vt.
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
Tempo Chemical Co., Inc., 47-02 5 St., Long Island City, 1, N.Y.
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.
Virginia Smelting Co., W. Norfolk, Va. (*p. 35*)
J. C. Whitlam Mfg. Co., Wadsworth, O.
X-Pando Corp., 43-15-36th St., Long Island City 1, N.Y.

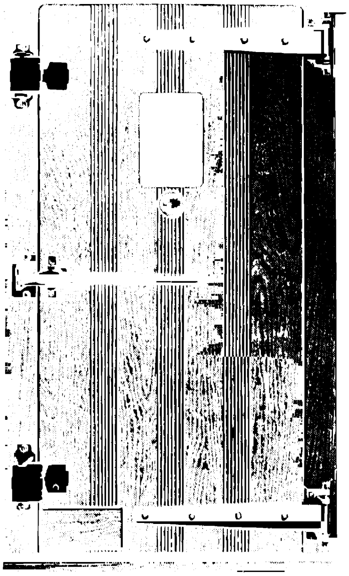
COMPOUNDS, MASTIC, (See COATINGS)

COMPRESSORS, AIR, GAS, etc.

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American-Marsh Pumps, Inc., 1948 Capital Ave., N.E., Battle Creek, Mich.
Binks Mfg. Co., 3114 Carroll Ave., Chicago 16, Ill. (*p. 73*)
Brunner Mfg. Co., Utica 1, N.Y. (*p. 48*)
Carrier Corp. 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Clark Bros Co., Div. of Dresser Industries, Olean, N.Y.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. (*p. 7*)
Cooper-Bessemer Corp., Mt. Vernon, O.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.
Fuller Co., Box 420, Catasauqua, Pa. (*p. 47*)
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Joy Mfg. Co., Oliver Bldg., Pittsburgh, Pa.
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Nash Engrg. Co., 358 Wilson Rd., S. Norwalk, Ct.
Niles Tool Wks. Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.
Pennsylvania Pump & Compressor Co., Easton, Pa.
Quincy Compressor Co., 217 Maine St., Quincy, Ill.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Union Steam Pump Co., Battle Creek, Mich.
United Engine Co., W. Holmes Rd., Lansing 12, Mich.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.

"Butcher Boy" COLD STORAGE DOORS

Since 1885



This "Butcher Boy" standard type cold storage door is equipped with "Super-Seal" fasteners and "Finger-Tip" Door Openers for greatest efficiency.

NOTE—

① Any "Butcher Boy" cold storage door can be hung either right hand or left hand at any time without change in construction as all "Butcher Boy" doors have full width hinge blocks

② All "Butcher Boy" doors maintain a 100% tight seal, and insure ease in opening and closing operations at all times because of exacting standards of construction

"Butcher Boy" NEW-TYPE HARDWARE can be installed on *any* cold storage door

● Super-Seal FASTENERS

"Super Seal" Fasteners maintain a 100% tight seal between the door gaskets and frame, thereby preventing WARPING and DOOR ICING

CONSTANT AUTOMATIC PRESSURE is brought against the door gaskets by two (2) Extra Heavy Duty fasteners

DOUBLE CONTACT locking feature of "Super Seal" replaces the old inefficient single contact fasteners, distributes even pressure upon the entire surface of the door and its gaskets

EASY TO INSTALL on any door ... adjustments are simple and positive with lock nut fittings

DURABLE, HEAVY, MODERN ... each Super Seal fastener weighs 10 lbs

● FINGER-TIP DOOR OPENERS

Minimize the efforts of opening heavy cold storage doors by application of the laws of "leverage".

Mere "finger tip" pressure will open any size door

Installed without drilling holes through the doors, eliminates the necessity of push rods which frequently stick, freeze or bend

Extra heavy, long, modernistic handles—designed for perfect operation on an 8 to 1 leverage ratio

"Finger Tip" openers snap back into closed position after each operation, eliminating the ever constant threat of tearing clothing or catching on a protruding handle

Write for Bulletins

VESTIBULE DOORS are used primarily to save refrigeration costs where long periods of opening prevail

"Butcher Boy" PRODUCTS

BUTCHER BOY COLD STORAGE DOOR CO.

170-180 N. SANGAMON ST. ★ CHICAGO 7, ILL. U. S. A.

COMPRESSORS, AMMONIA (See also CONDENSING UNITS, AMMONIA)

Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Clark Bros. Co., Inc., Olean, N.Y.
Cooper-Bessemer Corp., Mt. Vernon, O.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Fuller Co , Box 420, Catasauqua, Pa. *(p. 47)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.
York Corp., York, Pa. *(p. 52)*

COMPRESSORS, BOOSTER

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.

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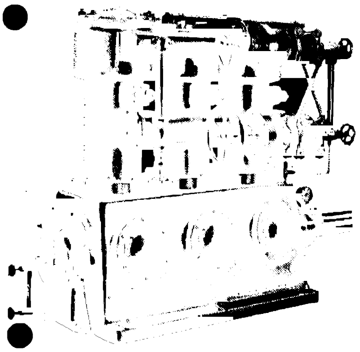

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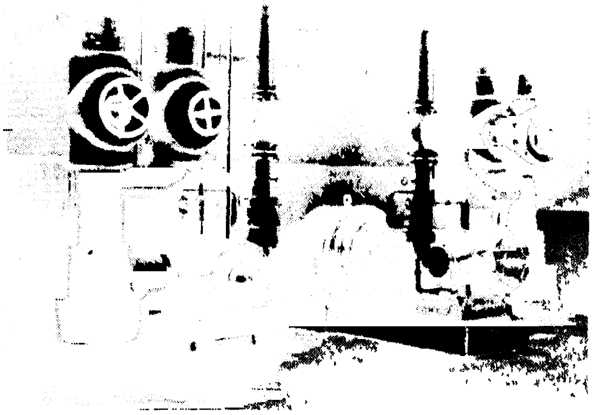

Two FES-FULLER ammonia booster compressors at Modesto, Calif. Average conditions: 5-in. vacuum to 30-lb. gage. 100-hp. motor on each compressor.

FES

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for low cost refrigeration

The FES-FULLER Rotary Compressor has been especially designed for use as a booster refrigeration compressor. It is suitable for pumping all types of refrigerants, including ammonia, from low-side conditions met in general practice, up to and including discharge pressures of 35 psig. Compressors are available in twenty-one sizes, with capacities ranging from 2 to 250 tons of refrigeration.

The efficient design of this equipment provides low-temperature refrigeration at low initial cost with economical operation. The compactness of design and flexibility of drive makes it possible to install these compressors practically anywhere . . . in fact, they have been installed in what was considered waste space in the machine room.

Some advantages are—direct or belt drive; accessibility; low maintenance; simple foundations. Blades automatically compensate for wear, maintaining capacity for the life of the machine. Oil-cooled jacket . . . oil-cooled double seal. Small floor space with relation to capacity. No bearing take-up; no gas-line pulsations. No leaking valves or grinding of seats. Eliminates a multiplicity of parts requiring frequent attention, adjustment, and replacement.

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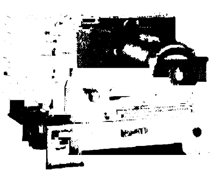
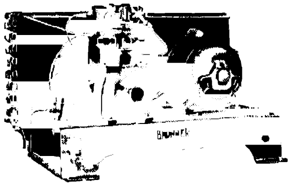
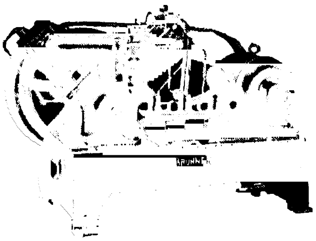
helps you serve better







69 Different Models, Types and Capacities of

REFRIGERATION CONDENSING UNITS

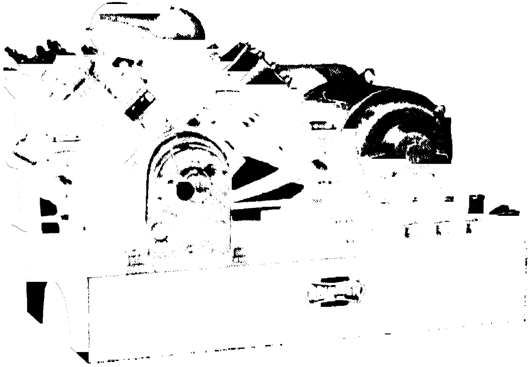
Air and Water Cooled

¼ HP. to 75 HP.

In the Brunner line you will find the exactly right unit to answer just about every commercial refrigeration or air conditioning application imaginable.

Brunner condensing units are well known for their outstanding performance and economy in operation. Specify and install Brunner's with confidence. Factory sales and engineering co-operation is yours for the asking. Write us. Catalogs on request.

BRUNNER
MANUFACTURING CO.
Utica 1, N. Y., U. S. A.



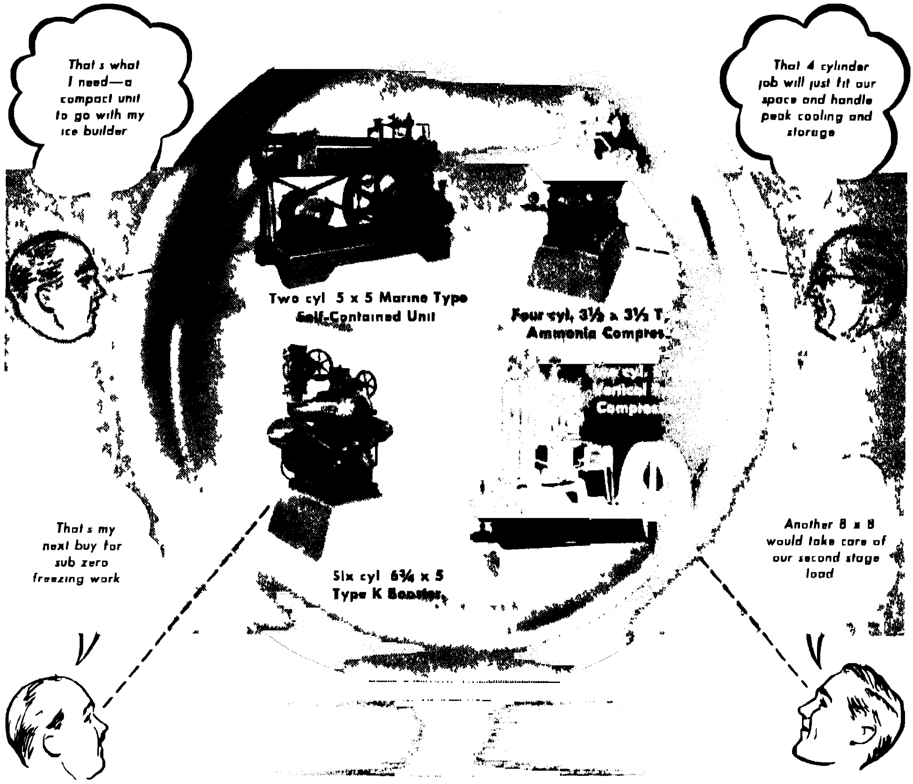
BRUNNER
SINCE 1906

Self Contained

AIR CONDITIONERS

4 SIZES — 3 - 5 - 7½ and 10 HP.

Remote Installation Types from 3 HP. to 75 HP.


That's what I need—a compact unit to go with my ice builder
That 4 cylinder job will just fit our space and handle peak cooling and storage
Two cyl 5 x 5 Marine Type Self-Contained Unit
Four cyl. 3½ x 3½ T. Ammonia Compres
cyl. Vertical Compres
That's my next buy for sub zero freezing work
Another 8 x 8 would take care of our second stage load
Six cyl 6¾ x 5 Type K Booster

Look ahead with CP Refrigeration

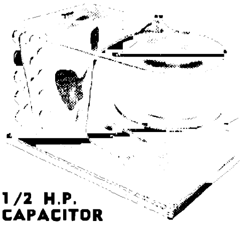
From the CP Balanced Line you'll get —not just a compressor—but *exactly* the unit *you* need, to do *your* refrigeration job large or small with efficiency and economy assured by CP's more than 60 years of refrigeration pioneering and leadership. You're *right* when you choose CP.

THE CREAMERY PACKAGE MFG. COMPANY

General and Export Offices: 1243 W. Washington Blvd., Chicago 7, Illinois

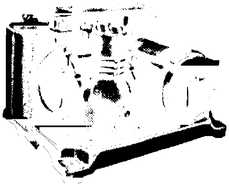
Lehigh BLU-COLD COMMERCIAL REFRIGERATION

A Complete Line ¼ to 5 HP


1/2 H.P. CAPACITOR

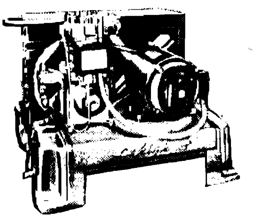
HERMETICS

Thoroughly proven design.
LOW TORQUE and CAPACITOR TYPES
1/4 thru 1/2 H.P.


1/2 H.P. PACKAGED

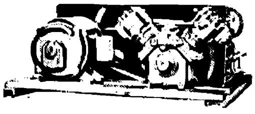
PACKAGED UNITS

Heavy duty capacity and construction. Exceptionally small mounting dimensions. Models for F-12 and F-22.
1/4 thru 1/2 H.P.


1/2 H.P. AIR COOLED

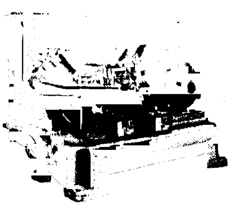
HEAVY DUTY UNITS

- AIR COOLED 1/3 thru 3 H.P.
- WATER COOLED 1/2 thru 5 H.P.
- AIR-WATER COOLED 1/2 thru 3 H.P.
- Units for Evaporative Condenser Application 1/2 thru 5 H.P.



TRUCK UNITS

- ALL-ELECTRIC for on-and-off road operation.
- UNITS FOR HOLD-OVER OPERATION . . . Compact units that provide more pay-load space. *3/4 thru 3 H.P.*



AUTOMATIC HIGH-SIDE DEFROST UNITS

For use with any evaporator.
3/4 thru 3 H.P.
—also available in truck models.

BARE COMPRESSORS
1/4 thru 5 H.P.

General Catalog On Request

Lehigh BLU-COLD
Lehigh Manufacturing Co.
PLANT LANCASTER, PENNA.

COMPRESSORS, BOOSTER (Continued)

Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Fuller Co., Catasauqua, Pa. *(p. 47)*
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Joy Mfg. Co., Oliver Bldg., Pittsburgh 22, Pa.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

COMPRESSORS, CARBON DIOXIDE

Frick Co., Waynesboro, Pa. *(p. 51)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Wittenmeier Machinery Co., 850 Spaulding N. Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*

COMPRESSORS, CENTRIFUGAL (See also CONDENSING UNITS, CENTRIFUGAL)

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Trane Co., La Crosse, Wis. *(p. 118)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*

COMPRESSORS, FREON, METHYL, etc. (See also CONDENSING UNITS)

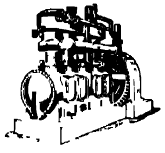
Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. *(p. 19)*
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Brunner Mfg. Co., Utica 1, N.Y. *(p. 48)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Copeland Refrigeration Corp., Sidney, O. *(p. 65)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Crider Corp., 2216 Northland, Lakewood, O.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis, 20, Mo. *(p. 7)*
Dieeler Div., General Machine & Mfg. Co., Blair St. & Spring Ave., Berwick, Pa.
Frick Co., Waynesboro, Pa. *(p. 51)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Lehigh Mfg. Co., Div. of Lehigh Foundries, Inc., 143 Fountain Ave., Lancaster, Pa. *(p. 50)*
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 53)*
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. *(p. 62)*
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Servel, Inc., Evansville 20, Ind.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 23)*
Tecumseh Products Co., Tecumseh, Mich. *(p. 61)*

(Continued)

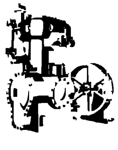
Frick Company

Waynesboro **Pennsylvania**

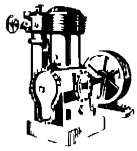
Branch Offices and Distributors in Principal Cities



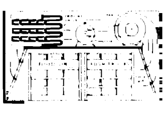
Large 4-Cylinder Compressors, for Ammo. or Freon-12



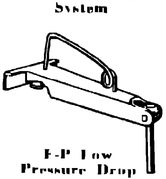
2-Cyl. Booster Compressor



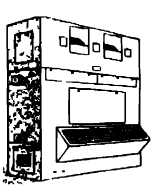
2-Cyl. Freon-12 Compressors



Section thru Frick-Freezing System



F-P Low Pressure Drop Tube & Bracket for Ice Plants



Evaporative Condenser

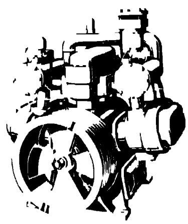
AMMONIA REFRIGERATION

Capacities up to 1000 tons in one unit. Compressors feature water jacket over cylinder heads, outboard bearing beyond wheel, force-feed oiling, and patented Flexo-Seals Capacity controls, dual pressure cylinders, etc., supplied on order. Booster compressors save power for low temperature Frick freezing systems. See Bulletins 104, 112, 516 and 651.

LOW-PRESSURE REFRIGERATION

Units from ¼ through 15 hp. Air and water cooled, for Freon-12 Separate machines in sizes 5¼″ x 4″ through 17¾″ x 12″, four cylinder.

The NEW "ECLIPSE" machines, 4⅜″ x 4¼″, give big capacity in small space, have many exclusive features: Flexo Seals, force-feed oiling, dynamic balancing, capacity controls with water cooled valve plates. Bul. 97 & 100.



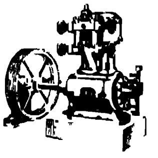
Frick NEW "ECLIPSE" Compressors have 2, 3, 4, 6 or 9 Cyl.

AIR CONDITIONING SYSTEMS

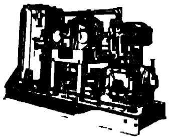
Complete systems, as well as unit conditioners, also refrigeration for use with equipment built by others Ask for Bulletins 502, 503, 504, and 505.

ICE-MAKING SYSTEMS

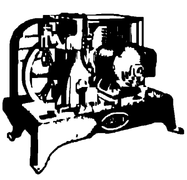
Loose can or group lift; brine race with Vertiflow unit or brine cooler systems; manual or automatic control One man systems up to 100 tons Bulletins 50, 51, 127 and 111



2-Cylinder Ammonia Compressors

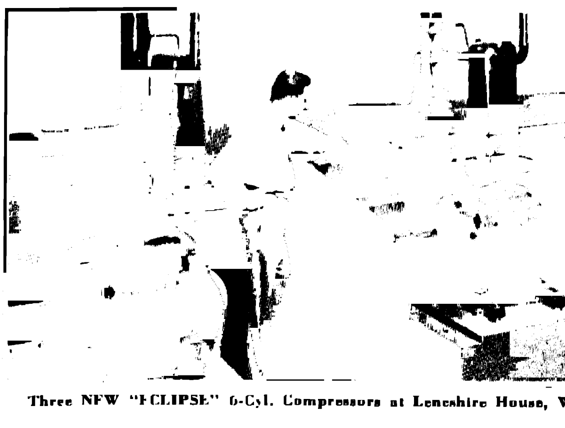


Combined Ammonia Unit



Low Pressure Unit

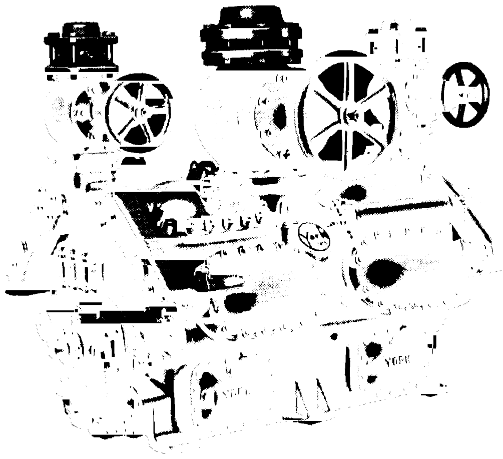
Frick Unit Air Conditioner



Three NEW "ECLIPSE" 6-Cyl. Compressors at Leneshire House, Washington, D.C.

THE NEWEST IN YORK V/W AMMONIA COMPRESSORS

NOW AVAILABLE IN
4, 6, 8, 12 AND 16
CYLINDERS



Only YORK Gives You All These Features:

1. LOW INITIAL COST.
2. LOW OPERATING COST.
3. LOW MAINTENANCE COST.
4. SPACE-SAVING COMPACTNESS
5. SIMPLIFIED MOTORS AND STARTING CONTROL.
6. VIBRATION-FREE OPERATION.
7. FILTERED LUBRICATION.
8. POSITIVE LUBRICATION.
9. POSITIVE SHAFT SEAL.
10. BOOSTER AND/OR HIGH STAGE OPERATION.
11. COMPLETE LINE.

The York V/W Compressor can't wear out because all parts subject to wear are renewable. And simplicity of design plus speed and ease of repair save you time and money consuming delays.

Work plans based on product-load schedule instead of fixed machine capacity. Product costs are reduced in just about straight line ratio with product load—only 28% power at 25% load.

More than 6,000 successful installations—both Freon and Ammonia—in operation in every type of industry twelve years or more, prove that V/W can match your every industrial processing need. For further proof, see your York Representative today. Or write direct to York Corporation, York, Penna.

York
refrigeration
air conditioning

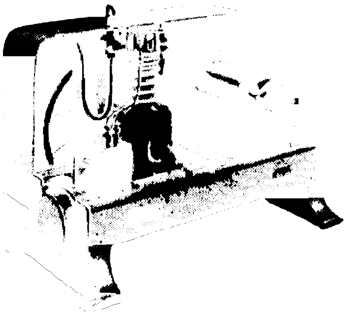
The big advances come from

YORK

Headquarters for **– Refrigeration and Air Conditioning**

MILLS COMPRESSORS AND CONDENSING UNITS

10 H.P. Down to ¼ H.P.

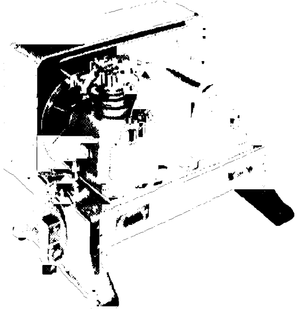


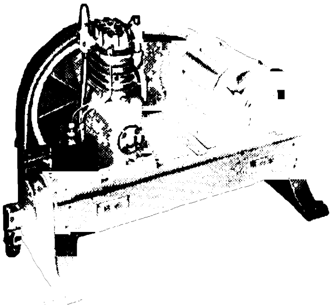
AIR-COOLED UNITS

H P.	WIDTH	DEPTH	HEIGHT	NET WGT
¼	26"	18"	17"	142 lbs
⅓	26"	18"	17"	146 lbs
½	30½'	21½"	19½"	200 lbs
¾	36½"	23"	24"	276 lbs
1	36½"	23"	24"	286 lbs
1½	42¾"	25½"	31½'	450 lbs
2	42¾"	25½"	31½"	460 lbs
3	42¾"	26½'	31½"	470 lbs

COMBINATION AIR- AND WATER-COOLED UNITS

H P	WIDTH	DEPTH	HEIGHT	NET WGT
½	33½"	21½"	19½"	212 lbs
¾	39"	23"	24"	292 lbs
1	39"	23"	24"	306 lbs
1½	47¼"	25½"	31½"	470 lbs
2	47¼"	25½"	31½"	482 lbs
3	53"	26½"	31½"	492 lbs



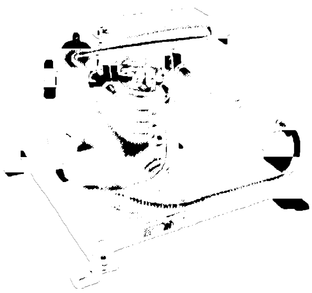


WATER-COOLED UNITS

H P	WIDTH	DEPTH	HEIGHT	NET WGT
⅓	33½"	17"	19½"	188 lbs
½	33½"	17"	19½"	194 lbs
¾	39"	18½"	24"	265 lbs
1	39"	18½"	24"	272 lbs
1½	47¼"	21¼"	29½"	435 lbs
2	47¼"	21¼ '	29½"	445 lbs
3	53"	21¼"	29½"	455 lbs
5	63"	29½"	32"	895 lbs
7½	63"	29½"	32"	945 lbs
10	63"	30"	34"	1050 lbs

CLOSE-COUPLED UNITS

H.P.	WIDTH	DEPTH	HEIGHT	NET WGT
¼	19¼"	16¾"	14¼"	106 lbs
⅓	19¾"	18"	14¼"	118 lbs.
½	22"	18"	14½"	159 lbs



There's a Mills For Every Installation

For complete information write for New Catalog 204-1

MILLS INDUSTRIES, Incorporated, 4100 Fullerton Ave., Chicago 39, Ill.

COMPRESSORS, FREON, METHYL (Continued)

Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y
Universal Cooler Div., Tecumseh Products Corp., 299 Joseph St., Marion, O. (*p. 61*)
U. S. Thermo Control Co., 44 S. 12th St., Minneapolis 4, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

COMPRESSOR PARTS, DRAWN OR STAMPED (See also STAMPINGS; also SHAPES, DRAWN; etc.)

Ackermann Mfg. Co., Wheeling, W.Va.
Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. (*p. 178*)
Bingham-Herbrand Corp., 1062 Post St., Toledo 6, O.
Chicago Seal Co., 332 S. Hoyne Ave., Chicago 20, Ill.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.
Industrial Wire Cloth Products Corp., Wayne, Mich. (*p. 88*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich.
Metal Specialty Co., Este Ave., & B&O R.R., Cin'ti., O
Transue & Williams Steel Forging Corp., Alliance, O.
Worthington Pump & Machinery Corp., Harrison, N.J.

CONCRETE, INSULATING (See INSULATING PLASTER & CONCRETE)

CONDENSATE DISPOSAL UNITS

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.

CONDENSER WATER DISTRIBUTORS

Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

CONDENSER WATER REGULATING VALVES

Amfnco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. (*p. 149*)
Art Valve Co., 4662 N. Lincoln Ave., Chicago 25, Ill.
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Baker Refrigeration Corp., Inc., S. Windham, Me. (*p. 63*)
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Electrimatic Div., Jas. P. Marsh Corp., 3501 Howard St., Skokie, Ill.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Penn Elec. Switch Co., Goshen, Ind. (*p. 69*)
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. (*p. 155*)
York Corp., York, Pa. (*p. 52*)

CONDENSERS, AIR COOLED

American Coils Co., 25 Lexington St., Newark 5, N.J.
Brown Fintube Co., Elyria, O.
Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. (*p. 40*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y. (*p. 59*)
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Houdaille-Hershey Corp., 1900 Foss Park Ave., North Chicago, Ill. (*p. 54*)
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 83*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Kirsch Co., Sturgis, Mich.
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. (*p. 55*)
McQuay Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. (*p. 200*)
Mills Industries, Inc., 4100 Fullerton Ave., Chicago 39, Ill. (*p. 53*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)
Rudy Mfg. Co., Dowagiac, Mich.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. (*p. 118*)
Universal Cooler Div., Tecumseh Products Corp., 299 Joseph St., Marion, O. (*p. 61*)
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)
Young Radiator Co., Racine, Wis. (*p. 77*)

CONDENSERS, ATMOSPHERIC

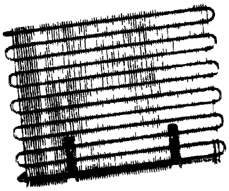
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. (*p. 118*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

PLATE-TYPE CONDENSERS

Mechanically bonded tube-plate condensers, high in efficiency and low in cost, are available in production quantities and in a broad range of sizes to meet your particular requirements for refrigerators and freezers.

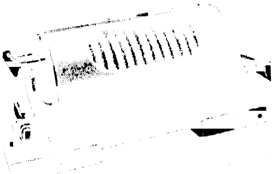
HOUDAILLE-HERSHEY CORP.
North Chicago Division, North Chicago, Illinois

McCORD Refrigeration and Air Conditioning Products



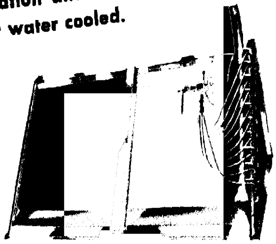
CONTINUOUS TUBE DOMESTIC CONDENSER

Round continuous tube, plate or fin type condensers for domestic refrigeration.



WATER COOLED CONDENSER

Condensers for commercial refrigeration and air conditioning, air or water cooled.



BACKED BY 42 YEARS of Heat Transfer Experience

A pioneer in the Heat Transfer Industry, many outstanding developments have come from McCord laboratories. Both spiral and extended fin surfaces are developments of McCord. McCord surfaces are designed to meet all heat transfer requirements at a minimum of cost. Positive contacts insure permanency of rated heat transfer capacity. Maintain your reputation by specifying McCord-engineered heat transfer surface.

McCORD CORPORATION
Detroit 11, Michigan

AIR CONDITIONING EVAPORATOR

Evaporator coils for air conditioning, engineered to meet your heat transfer requirements.

CONDENSERS, CARBON DIOXIDE

Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

CONDENSERS, COMBINATION AIR & WATER COOLED

Baltimore Aircoil Co., Inc., 2519 Pennsylvania Ave., Baltimore 17, Md (*p. 95*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25,*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33) Cal.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. (*p. 74*)
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. (*p. 53*)
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles 22, Cal. (*p. 196*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago 20, Ill.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Trane Co., La Crosse, Wis. (*p. 118*)
York Corp., York, Pa. (*p. 52*)

CONDENSERS, DOUBLE PIPE

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
L. O. Koven & Bro., Inc., Jersey City 7, N.J.
Nooter Corp., 1426 S. 2nd St., St. Louis 4. Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p 52*)

CONDENSERS, DOUBLE TUBE

Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Refrigeration Div. Yates-American Machine Co., Beloit, Wis.
Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa. (Cleanable) (*p. 57*)
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 88*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. (*p. 55*)
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Reese & Long Refrigeration Products, Inc., 1891 Park Ave., N.Y.C. 35
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)
Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago 12, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

CONDENSERS, ELECTRIC (See CAPACITORS)

CONDENSERS, EVAPORATIVE (See EVAPORATIVE CONDENSERS)

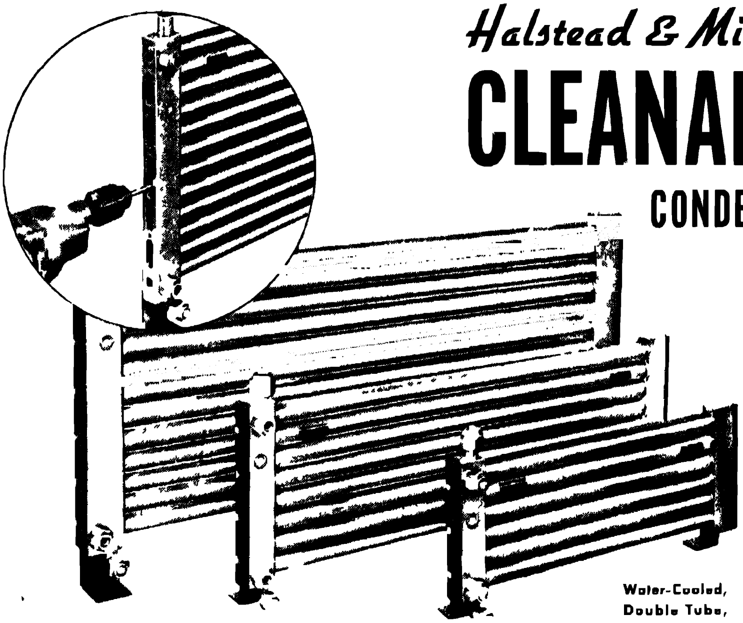
CONDENSERS, SHELL & COIL

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Croll Reynolds Engrg. Co. Inc., 17 John St. N.Y.C.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. (*p. 7*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
John J. Dupps Co., Germantown, O.
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City 7, N.J.
Kramer Trenton Co., Olden & Bruening Aves., Trenton 5, N.J. (*p. 195*)
Lehigh Mfg. Co., Lancaster, Pa. (*p. 59*)
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. (*p. 55*)
Nevinger Mfg. Co., Greenville, Ill.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)

(Continued)

IN SERVICE
IN QUALITY

Halstead & Mitchell
CLEANABLE
CONDENSERS



Water-Cooled,
Double Tube,
Counter-flow

Regardless of whose condensing unit you buy in any size from ⅓ through 25 H.P.—your first demand is to insist that it have a CLEANABLE water-cooled condenser. It costs no more originally ... and the added advantages in longer life and more efficient service can't be overlooked. No matter how poor the water conditions and no matter how long the service of the unit, you can always count on restoring new-unit efficiency by the simple use of a spiral cleaning tool which will thoroughly remove *mechanically* all the harmful corrosive material that might accumulate on the water tube interiors. You needn't settle for less, for almost all leading manufacturers are recognizing the advantages of cleanability—and are equipping their units accordingly, in *all* sizes.

Capacity Range ⅓ through 25-ton.

JOBBERS in all principal cities carry HM condensers in stock for immediate delivery

SEND FOR DESCRIPTIVE LITERATURE

HM

Halstead & Mitchell

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Chicago 4
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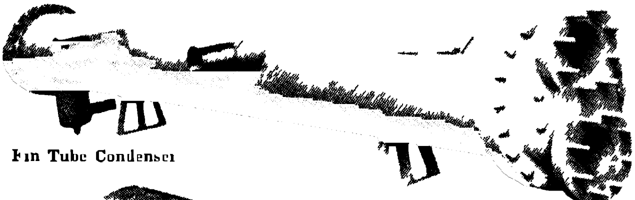
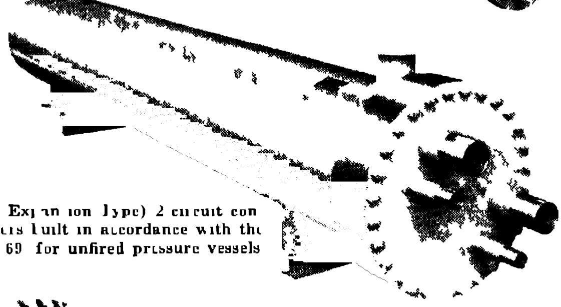
p·k Heat Exchangers for Heating, Air Conditioning and Refrigeration Service Hot Water Storage Heaters — Instantaneous Heaters — Converters — Fuel Oil Heaters — Freon Coolers — Condensers — Balance Loaders — Interchangers — Suction Line Exchangers — Slug Eliminators — Coolers for Water, Brine and other Liquids

Since 1880, **p·k** has designed, engineered and fabricated heat transfer equipment to meet industries' most exacting requirements. We are prepared to recommend and design exchangers for air conditioning, refrigeration, process and industrial cooling applications

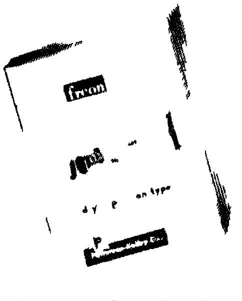


p·k Freon 12 Fin Tube Condenser



p·k Freon Cooler (Dry Expansion Type) 2 circuit construction All **p·k** Coolers built in accordance with the A S M E Code Par U 69 for unfired pressure vessels

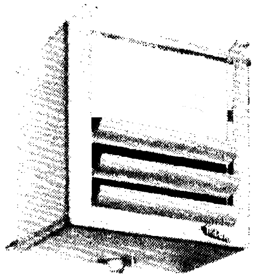
p·k Freon Cooler (Dry Expansion Type) 3 circuit construction for 200-ton refrigeration capacity



Catalog number 101 contains tables charts and complete data which will enable you to select the Cooler which exactly meets your needs

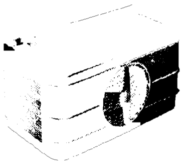
fedders

ENGINEERING AND PRODUCTION SERVICE TO MANUFACTURERS, JOBBERS, DISTRIBUTORS, DEALERS AND SERVICE MEN



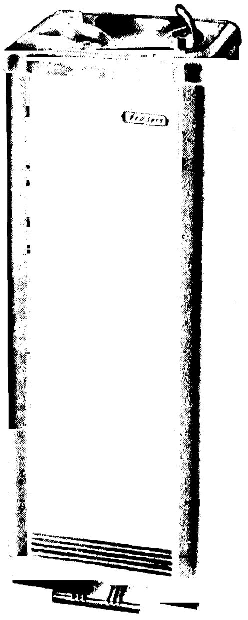
UNIT COOLERS

Suspended and panel types featuring rust proof cabinets, high efficiency, thoroughly dehydrated cooling elements and quiet fans. Heat exchanger on all but small sizes.



AIR CONDITIONERS

Window and console types . . . ⅓ ton to 1½ ton.

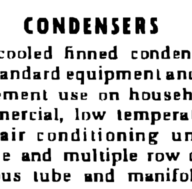


WATER COOLERS

Bottle, bubbler, air cooled, water cooled and explosion-proof models.

SUPER HEAT THERMOMETERS

For accurate setting of controls of refrigeration and air conditioning systems.



CONDENSERS

Air cooled finned condensers for standard equipment and replacement use on household, commercial, low temperature and air conditioning units. Single and multiple row continuous tube and manifolded types.

Write for Bulletins on Fedders Products

FEDDERS-QUIGAN

CORPORATION

57 TONAWANDA STREET • BUFFALO 7, NEW YORK

CONDENSERS, SHELL & COIL (Continued)

Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago 12, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
B. F. Sturtevant Div., Westinghouse Elec. Co., 101 Readville St., Boston 36, Mass. (*p. 23*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15 N.Y.
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

CONDENSERS, SHELL & TUBE

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Baker Refrigeration Corp., Inc., S. Windham, Me. (*p. 63*)
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Croll-Reynolds Engrg. Co., 17 John St., N.Y.C.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. (*p. 7*)
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 81*)
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
John J. Dupps Co., Germantown, O.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.H. (*p. 29*)
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Kramer-Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Lummus Co., 420 Lexington Ave., N.Y.C.
National Refrigerators Co., 827 Koeln Ave., St. Louis, Mo.
Nevinger Mfg. Co., Greenville, Ill.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave. Chicago 18, Ill.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago 12, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. (*p. 118*)
U. S. Air Conditioning Corp., Como Ave. S.E. at 33rd, Minneapolis 14, Minn. (*p. 37*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7 Wis. (*p. 33*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Westinghouse Elec. Corp., P.O. Box 7348, S. Philadelphia 1, Pa.
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

CONDENSING UNITS, AIR COOLED, FREON & METHYL

Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. (*p. 19*)
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Blissfield Mfg. Co., 626 Depot St., Blissfield, Mich.
Brunner Mfg. Co., Utica 1, N.Y. (*p. 48*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Copeland Refrigeration Corp., Sidney, O. (*p. 65*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chidago 7, Ill. (*p. 49*)
Crider Corp., 2216 Northland, Lakewood, O.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. (*p. 7*)
Dieeler Div., General Machine & Mfg. Co., Blair St. & Spring Garden Ave., Berwick, Pa.
Frick Co., Waynesboro, Pa. (*p. 51*)
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 83*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Lehigh Mfg. Co., Div. of Lehigh Foundries, Inc., 143 Fountain Ave., Lancaster, Pa. (*p. 50*)
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. (*p. 53*)
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Servel, Inc., Evansville 20, Ind.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Tecumseh Products Co., Tecumseh, Mich. (*p. 61*)
Trane Co., La Crosse, Wis. (*p. 118*)
Universal Cooler Div., Tecumseh Products Corp., 299 Joseph St., Marion, O. (*p. 61*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

CONDENSING UNITS, AMMONIA

Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Creamery Package Mfg. Co. 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.

(Continued)

TWO OUTSTANDING NAMES

Now combined to offer the refrigeration industry the most complete line of compressors and condensing units available.

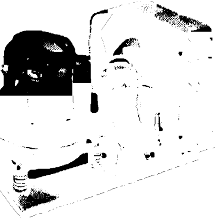
(Full range of sizes 1/9—15 h.p.)

★ TECUMSEH PRODUCTS COMPANY

Plant at Tecumseh, Michigan manufacturing hermetic units from 1/9 H.P. to 3/4 H.P., conventional units from 1/6 H.P. to 1/2 H.P. and conventional compressors from 1/6 H.P. to 1 H.P.

single compressor

1/9 h.p. compressor

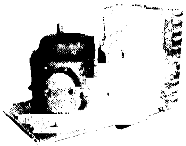


1/3 h.p. condensing unit

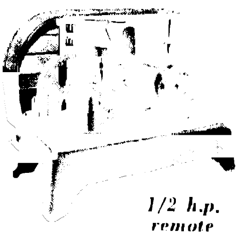
The Tecumseh line of hermetics has long been known in the refrigeration industry for dependable, trouble-free performance. Over 8 million Tecumseh compressors are now in use in the field in freezers, household refrigerators, beverage coolers, water coolers, etc. Three out of four compressors, condensing units and systems purchased by manufacturers in 1950 for incorporation in refrigerating equipment were built by Tecumseh.

★ UNIVERSAL COOLER DIVISION

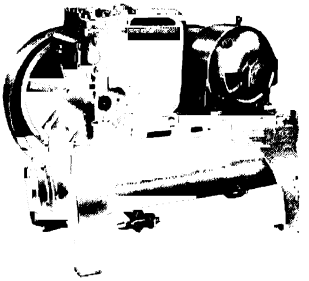
Plant at Marion, Ohio manufacturing air cooled units in all sizes up to 3 H.P., remote units from 1/2 H.P. to 15 H.P., hermetics from 1/8 H.P. to 1-1/2 H.P.



1/5 h.p. close coupled



1/2 h.p. remote base



15 h.p. water cooled

The Universal Cooler line of compressors and condensing units has been serving the refrigeration industry for 29 years. With a full range of sizes up to 15 H.P., the Universal Cooler Division can supply a condensing unit for any application.

★ This combination of two outstanding names in the refrigeration industry is able to offer manufacturers and jobbers the most complete line, with respect to sizes and models, of conventional compressors and condensing units of any line in the industry. ★

Complete data on these Tecumseh and Universal Cooler units is available on request. Write today for specifications on the type and size unit you require in your particular application.

Dept. A-1
TECUMSEH PRODUCTS COMPANY
TECUMSEH, MICH.

Dept. A-2
UNIVERSAL COOLER DIVISION
MARION, OHIO

EXPORT DEPARTMENT: 2111 WOODWARD AVE., DETROIT 1, MICH.

CONDENSING UNITS, AMMONIA (Continued)

Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Frick Co., Waynesboro, Pa. (*p. 51*)
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

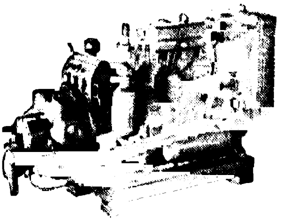
CONDENSING UNITS, CENTRIFUGAL

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Trane Co., La Crosse, Wis. (*p. 118*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

CONDENSING UNITS, GASOLINE DRIVEN

Brunner Mfg. Co., Utica 1, N.Y. (*p. 48*)

READY-POWER
ENGINED REFRIGERATION
for Lowest Cost Air Conditioning and Refrigeration



Powered by Industrial Engines Using *Gasoline* *Natural Gas* *Butane or* *Diesel Fuel*

For dependable air conditioning or refrigeration anywhere at lowest cost! Ideal for stationary, portable or marine operation.

THE READY-POWER CO.
11231 FREUD AVE., DETROIT 14, MICHIGAN.

Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. (*p. 7*)
Lehigh Mfg. Co., Lancaster, Pa. (*p. 50*)
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. (*p. 62*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.
Universal Cooler Div., Tecumseh Products Corp., 299 Joseph St., Marion, O. (*p. 61*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

CONDENSING UNITS, HERMETIC OR SEMI-HERMETIC, FREON & METHYL

Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. (*p. 19*)
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Copeland Refrigeration Corp., Sydney, O. (*p. 65*)
Crider Corp., 2216 Northland, Lakewood, O.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Lehigh Mfg. Co., Lancaster, Pa. (*p. 50*)
Lynch Corp., 3600 Summit St., Toledo 1, O.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. (*p. 53*)
Servel, Inc., Evansville 20, Ind.
B. F. Sturtevant Div., Westinghouse Elec. Co., 101 Readville St., Boston 36, Mass. (*p. 23*)
Tecumseh Products Co., Tecumseh, Mich. (*p. 61*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
Universal Cooler Div., Tecumseh Products Div., 299 Joseph St., Marion, O. (*p. 61*)
York Corp., York, Pa. (*p. 52*)

CONDENSING UNITS, HOUSEHOLD

Blissfield Mfg. Co., 626 Depot St., Blissfield, Mich.
Copeland Refrigeration Corp., Sidney, O. (*p. 65*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. (*p. 53*)
Servel, Inc., Evansville, Ind.
Tecumseh Products Co., Tecumseh, Mich. (*p. 61*)
Universal Cooler Div., Tecumseh Products Corp., 299 Joseph St., Marion, O. (*p. 61*)

CONDENSING UNITS, PROPANE

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Croll-Reynolds Engrg. Co., Inc., 17 John St., N.Y.C.
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlein Ave., St. Louis 20, Mo. (*p. 7*)
Propane Development Corp., 41 Murray St., N.Y.C. 7
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. (*p. 62*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

CONDENSING UNITS, WATER COOLED, FREON & METHYL

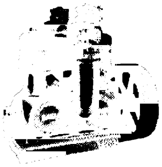
Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. (*p. 19*)
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Brunner Mfg. Co., Utica 1, N.Y. (*p. 48*)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Copeland Refrigeration Corp., Sidney, O. (*p. 65*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. (*p. 7*)

(Continued)

BAKER REFRIGERATION CORPORATION

Air Conditioning • Refrigeration • Ammonia • Freon

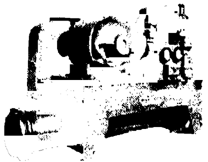
FREON



Baker Compressors, 1½ to 60 H.P., feature high efficiency at low operating cost. Timken bearings, forced feed lubrication, gear-type oil pump, positive shaft seal promote long life.

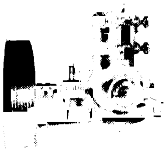
The Baker MRU® Line

Baker MRU equipment provides over 1000 designs to match the machine exactly to requirements. From 5 to 60 H.P. with capacity control for use with shell and tube or evaporative condenser, or water tower.

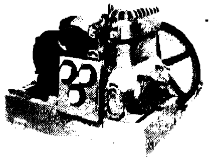


Baker Condensing Units, ⅓ to 60 H.P., are engineered with matched components to assure efficient, trouble-free operation for every application. Compressor Units, 1½ to 60 H.P.

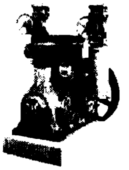
AMMONIA



Baker Ammonia Compressors, 2 to 125 H.P., can be supplied with capacity reduction control down to 33⅓%. Double suction control for operation at two different temperatures is available on special order.

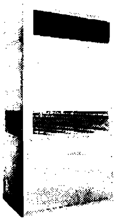


Baker Compressor Units, 2 to 15 H.P., feature automatic motor control with overload and low voltage protection. Complete Condensing Units are also available from 2 to 15 ton capacities, ready for installation.



Baker Booster Compressors for two-stage operation on the low side in low temperature applications. Forced-feed lubrication and balanced fly wheel for "V" belt drive. Compact design requires a minimum of space.

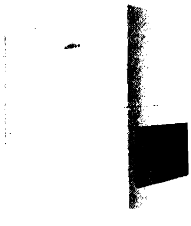
BAKERAIRE



Delivers maximum cooling per H.P. input. Famous Baker open-type or hermetic compressor in 3, 5, 7½ and 10 ton capacities. 4-way air discharge, new Sphericoil condenser, humidity control, spring-mounted compressor.

CENTRAL-AIR

A complete unit air conditioner in capacities from 5 to 40 tons with built-in evaporative condenser. Factory assembled and tested, delivered ready to operate. Simplified installation keeps first cost low. Fully automatic controls.



"Intentionally better since 1905"

PLUS

Condensers—evaporative or shell and tube, wide capacity range.

Liquid Receivers—for ammonia or Freon with positive liquid seal.

Cooling Units—for every requirement, for all refrigerants.

Valves and Fittings—a complete line for all applications.

Baker Refrigeration Corporation, South Windham, Maine

Baker

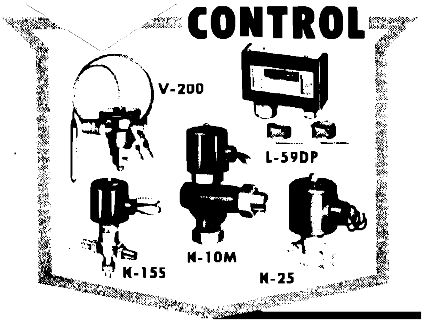
IR CONDITIONING
and REFRIGERATION

Offices, warehouses and parts depots in principal cities.

CONDENSING UNITS, WATER COOLED (Continued)

Diceler Div., General Machine & Mfg. Co., Blair St., & Spring Garden Ave., Berwick, Pa.
Frick Co., Waynesboro, Pa. *(p. 51)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Kohlenberger Engrg. Co., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Lehigh Mfg. Co., Div. of Lehigh Foundries, Inc., 143 Fountain Ave., Lancaster, Pa. *(p. 50)*
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 53)*
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. *(p. 62)*
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Servel, Inc., Evansville 20, Ind.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 23)*
Trane Co., La Crosse, Wis. *(p. 118)*
Universal Cooler Div., Tecumseh Products Corp., 299 Joseph St., Marion, O. *(p. 61)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
XL Refrigerating Co., 1834 W 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

1st choice for AUTOMATIC CONTROL



GENERAL CONTROLS
801 ALLEN AVENUE GLENDALE 1, CALIFORNIA

Manufacturers of Automatic Pressure, Temperature, Level and Flow Controls

FACTORY BRANCHES: Baltimore 5, Birmingham 3, Boston 16, Buffalo 3, Chicago 5, Cincinnati 2, Cleveland 15, Dallas 2, Denver 4, Detroit 21, Glendale 1, Houston 6, Indianapolis 5, Kansas City 2, Minneapolis 2, Newark 6, New York 17, Philadelphia 23, Pittsburgh 22, St. Louis 3, San Francisco 7, Seattle 1, Tulsa 6, Washington 6.
DISTRIBUTORS IN PRINCIPAL CITIES

CONDUIT, ELECTRICAL

Aluminum Co. of America, Pittsburgh 19, Pa.
American Brass Co., Waterbury 20, Ct. *(p. 189)*
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Chicago Metal Hose Corp., Maywood, Ill.
Cornish Wire Co., Rutland, Vt.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
General Electric Co., 1 River Road, Schenectady 5, N.Y.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
National Elec. Products Corp. Chamber of Commerce Building, Pittsburgh 19, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Triangle Conduit & Cable Co., Inc., New Brunswick, N.J.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Youngstown Sheet & Tube Co., Youngstown, O

CONDUIT SYSTEMS, INSULATED

Durant Insulated Pipe Co., 1015 Runnymeade St., E. Palo Alto, Cal.
Ric-Wil Co., Union Commerce Bldg., Cleveland 14, O.

CONNECTING RODS

Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
J. H. Williams & Co., 400 Vulcan St., Buffalo 7, N.Y.

CONNECTIONS, FLEXIBLE (See FLEXIBLE CONNECTIONS)

CONNECTORS, WIRE

Buchanan Elec'l Prod. Corp., 225 Route 29, Hillside N.J.
Burke Elec. Co., Erie, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cornish Wire Co., Rutland, Vt.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y
Ideal Industries, Inc., Sycamore, Ill.
Russell & Stoll Co., Inc., 125 Barclay St., N.Y.C.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Soreng Mfg. Corp., 9555 Eden Ave., Schiller Park, Ill.

CONTACTS, ELECTRICAL

Baker & Co., Inc., 113 Astor St., Newark 5, N.J.
Fansteel Metallurgical Corp., North Chicago, Ill.
General Control Co., 1200 Soldiers Field Rd., Boston 34, Mass.
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
General Plate Div., Metals & Controls Corp., Attleboro, Mass.
Gibson Elec. Co., 8350 Frankstown Ave., Pittsburgh 21, Pa.
Handy & Harman, 82 Fulton St., N.Y.C. 7
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls, Ind.
National Carbon Co., Inc., Unit. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Westinghouse Elec. Corp., Beaver, Pa.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

CONTINUOUS FREEZERS (See FREEZERS)

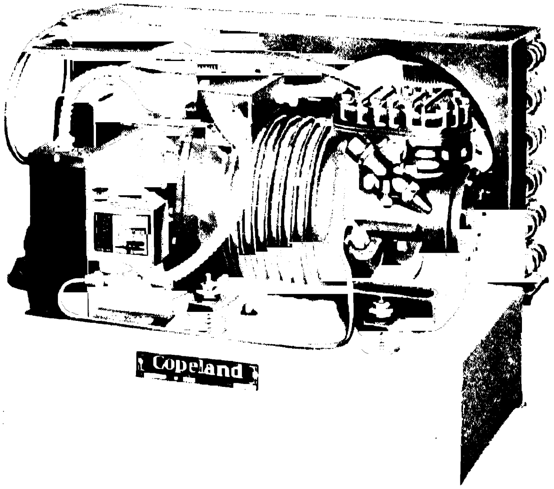
CONTROL SYSTEMS FOR AIR CONDITIONING & REFRIGERATION

Bahnson Co., 1001 S. Marshall St., Winston-Salem, N.C.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.

(Continued)

Copeland

DEPENDABLE *Electric* REFRIGERATION



QUIET

EFFICIENT

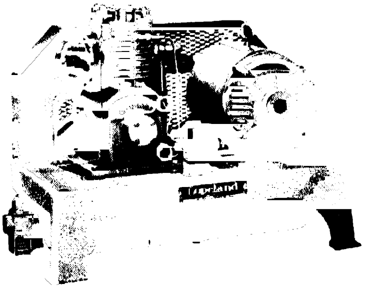
ECONOMICAL

RUGGED

COPELAMETIC THE ACCESSIBLE HERMETIC

AVAILABLE UP TO 7½ H.P.

Proved dependable in more than 1,000,000 installations of all types, Copelametic combines all the good points of open-type and welded-in units. It's completely accessible in the field for adjustment and replacement of parts. Does away with belts, seals, manual oiling; provides extra economy and protection. Copelametic units can be used on any application now served by open units. 1/6 H.P. to 7½ H.P. inclusive.



COPELAND OPEN-TYPE UNITS

There is a Copeland open-type unit in the correct size and design for any refrigeration use. Each unit is sturdy, compact. Air-cooled models from ¼ H.P. to 2 H.P. inclusive. Water-cooled models from ⅓ H.P. to 7½ H.P. inclusive. For self-contained installation, there are compact models from 1/6 to ½ H.P.

Manufacturers of: Refrigeration Units (Open-Type and Copelametic), Water Coolers, Refrigerators.

COPELAND REFRIGERATION CORPORATION • SIDNEY, OHIO

CONTROL SYSTEMS (Continued)

Fulton Sylphon Div., Robertshaw-Fulton Controls Co., Knoxville 4, Tenn.
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O.

CONTROLS, DEFROSTING CYCLE (See also CONTROLS PROGRAMS; also TIMERS)

Automatic Elec. Mfg. Co., 10 State St., Mankato, Minn.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
International Register Co., 2620 W. Washington Blvd., Chicago 12, Ill.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 63)*
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 68)*
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLS, HIGH PRESSURE OR HIGH PRESSURE CUTOUT

(A—Ammonia; B—Other refrigerants)

(B) Allen-Bradley Co., Milwaukee 4, Wis.
(A,B) Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
(B) Cutler-Hammer Inc., 315 N. 12th St., Milwaukee 1, Wis.
(A,B) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
(B) Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
(A,B) Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
(B) Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
(A,B) Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
(B) Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 68)*
(A,B) Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(A,B) U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
(B) White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.
York Corp., York, Pa. *(p. 52)*

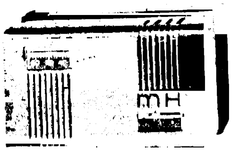
CONTROLS, HUMIDITY

Abbeon Supply Co., 58-10 41st Dr., Woodside, N.Y.
American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4. Md.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Supreme Elec. Products Co., 194 Vassar St., Rochester 7, N.Y.
Tagliabue Instruments Div, Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Walton Laboratories, Inc., 1186 Grove St., Irvington 11, N.J.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo

CONTROLS, LIQUID LEVEL (See also FLOAT SWITCHES; also HIGHSIDE FLOATS; also LOWSIDE FLOATS, etc.)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., P.O. Box 1342 Providence 1, R.I.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Climax Controls Div. of Black, Sivalls & Bryson, Inc., 15 N. Cincinnati, Tulsa, Okla.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Fischer & Porter Co., Hatboro, Pa.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Foxboro Co., 38 Newponset Ave., Foxboro, Mass.
Frick Co., Waynesboro, Pa. *(p. 51)*
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
Hays Corp., Michigan City, Ind.
Infilco, Inc., 2750 S. 12th Ave, Tucson, Ariz.
Jo-Bell Products, Inc., 4840 South St. Louis, Chicago, Ill. 15 N. Cincinnati, Tulsa, Okla.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Liquidepth Indicators, Inc., 42-26-28th St., Long Island City 1, N.Y.
McDonnell & Miller, Inc., 3500 N. Spaulding Ave., Chicago 18, Ill.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Soreng Mfg. Corp., 9555 Eden Ave., Schiller Park, Ill.
Tagliabue Instruments Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p. 89)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
York Corp., York, Pa. *(p. 52)*

See Honeywell FOR THE COMPLETE LINE OF AUTOMATIC CONTROLS THAT SERVES YOU BEST

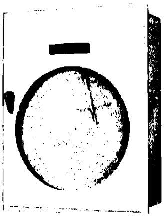


Frigistat

Minneapolis-Honeywell, with more than 65 years of experience, is best prepared to fill your control needs. Manufacturers of recorders, indicators, controllers, and special control panels for commercial and industrial refrigeration.

Frigistat

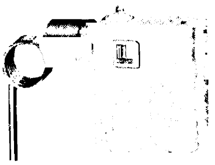
TA420A—the new horizontal type, snap acting thermostat for room temperature control of refrigerating systems. Horizontal style permits greater circulation of air around the sensitive vapor filled bellows. Modern, attractive cover has easy to read thermometer and harmonizing scale for reading temperature setting. T420A the mercury switch model is also available.



Recording Locker Room Thermometer

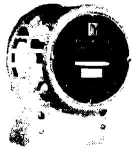
Recording Locker Room Thermometer

Model No. 60ZX1D-X-87HN; Chart No. 12777, Range —30°F. to +70°F., reverse. 12" diameter, seven-day revolution with one-hour time divisions. Has hand-wound chart drive as standard installation. Electric clock drive optional—self-starting synchronous electric clock, 115/230 v., 20-50-60 cycles. Equipped with 15' of bronze armored tubing and stainless steel bulb with 10" extension neck.



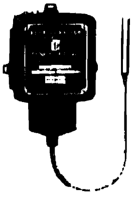
Temperature Controllers

L480 Remote Bulb Temperature Controller. Standard features include tamperproof shield for external adjustment dial; cold control; adjustable differential; extra terminal for reverse action; accurately calibrated dial.



Air Switch

T491A—Heavy Duty wall mounted thermostat. Mercury Switch contacts are actuated by a bi-metal element which is designed to withstand extreme conditions of temperature and humidity.



Temperature Controllers

Heavy duty remote bulb temperature controllers in a variety of ranges, with external scale plate marked high-low and calibrated at mid-scale. T414A for 2-wire switching; T614A for single pole-double throw switching for control of two speed motors, valves, alarm circuits, etc.

Pressure Controllers

L414A—Heavy Duty Suction Pressure Controller for use with non-corrosive refrigerants only. Direct reading range and differential scales.

Write for complete information on these and our many additional Refrigeration Controllers and accessories

MINNEAPOLIS Honeywell
First in Controls

Branch Offices in all principal cities

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Chicago, Illinois	**Philadelphia, Pa.**

Executive Offices
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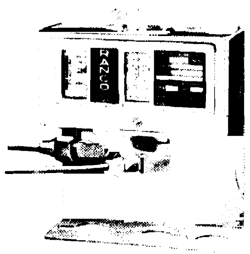
RANCO INC.

Columbus 1, Ohio, U.S.A.

Ranco Refrigeration Controls, designed and manufactured by refrigeration specialists, provide dependable, trouble-free service on the most exacting applications. The refrigeration industry uses more Ranco Controls than any other make. Check with Ranco Inc. first on your refrigeration control requirements.

Type "O" Refrigeration Controls

THE LEADING CONTROLS FOR FOOD PRESERVATION

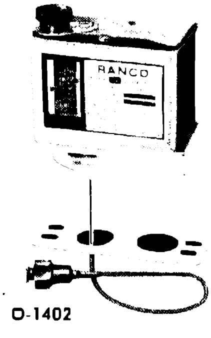

O-1505

O-1505—DUAL PRESSURE COMMERCIAL CONTROL. High pressure cut-out independently adjustable. Single pole, snap acting switch closes circuit on increase and opens circuit on decrease in pressure.

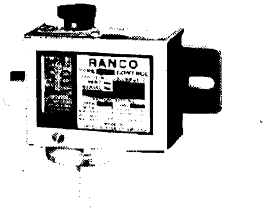
O-1401—LOW PRESSURE COMMERCIAL CONTROL. Graduated visible scale with calibrations for range and differential setting. Range screw changes cut-out and cut-in together. Differential screw changes cut-out only.

O-1401

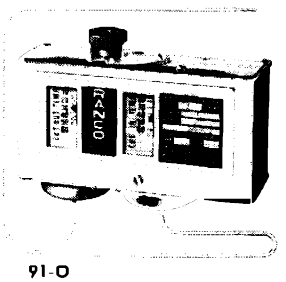
O-1402—LOW PRESSURE COMMERCIAL CONTROL. Similar to O-1401, but has constant cut-in. Like all Ranco type O Controls, is compact, sturdily constructed and adaptable to individual installation requirements.


O-1402

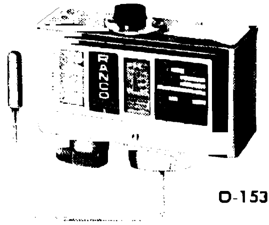
O-1419—TEMPERATURE COMMERCIAL CONTROL. Outside range and differential (cut-out) adjustments with graduated visible scales and calibrations. Lowest cut-out: F.; highest cut-in: 15° F.


O-1419

TYPE 91-O—EXCLUSIVE INTERLOCKING TWO-TEMPERATURE CONTROL. Assures uniform temperature, uniform high relative humidity and completely automatic defrosting of coil regardless of weather or load conditions or cold location of the compressor.


91-O

O-1535—DUAL TEMPERATURE COMMERCIAL CONTROL. Range screw changes temperature cut-out and cut-in together. High pressure cut-out independently adjustable. Differential screw changes temperature cut-out only.


O-1535

Ranco Inc.

RANCO CONTROLS

Available Through Leading Wholesalers

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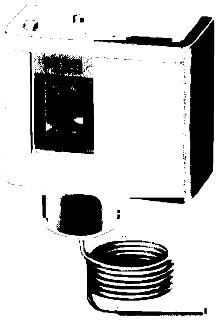
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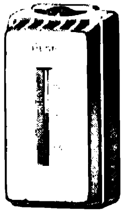
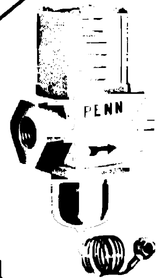

2

3

TRY 'EM!

then you'll use them on every refrigeration job



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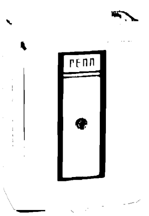


5

There is a difference in automatic controls . . . a difference in the degree of their efficiency, dependability and long-life accuracy! And once you try PENN controls . . . you'll learn that their performance on the job is the strongest recommendation for using PENN on every air conditioning and refrigeration application.

Take the first step in trying these better controls. Get the *new* PENN Condensed Catalog on Refrigeration Controls . . . it's FREE. Write *Penn Electric Switch Co., Goshen, Ind.* Export Division: 13 E. 40th Street, New York 16, U.S.A. In Canada: Penn Controls, Ltd., Toronto, Ont.

(1) Penn 270 Series temperature and pressure controls incorporate a direct-reading calibrated scale which shows both cut-in and cut-out points.

(2) Penn cooling thermostat for low or line voltage pilot service; two-wire, snap-acting. Humidistats also available.

(3) Penn Series 221 Solenoid Valves are direct acting and may be used with all non-corrosive refrigerants as well as with water, oil or air.

(4) Series 246 Water Valves are built in threaded and flanged styles for all refrigerants . . . sizes from $3/8$" to $2\frac{1}{2}$".

(5) Penn Magnetic Line Starters are built in NEMA sizes 0, 1 and $1\frac{1}{2}$ and are available as open-type models for control panels or with General Purpose enclosures.

PENN

AUTOMATIC CONTROLS

FOR HEATING, REFRIGERATION, AIR CONDITIONING, PUMPS, AIR COMPRESSORS, ENGINES, GAS RANGES

CONTROLS, MOTOR (See STARTERS)

CONTROLS, PROGRAM, SELECTIVE, SEQUENCE, etc. (See also TIME SWITCHES)

American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
R. W. Cramer Co., Inc., Centerbrook, Ct.
Fischer & Porter Co., Hatboro, Pa.
General Control Co., 1200 Soldiers Field Rd., Boston 34, Mass.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford, Ct.
Haydon Mfg. Co., 245 East Elm St., Torrington, Ct.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Sarco Co., Inc., 350 5th Ave., N.Y.C. 1 (*p. 167*)
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Wheelco Instruments Co., Harrison & Peoria Sts., Chicago 7, Ill.
White-Rodgers Electric Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLS, SUCTION PRESSURE ACTUATED

(A—Ammonia; B—Other refrigerants)
(a—Close on rise; b—With H.P. cutout; c—Reverse acting; d—With overload protection; e—Multiple circuit)

(B:a,b,e) Allen Bradley Co., Milwaukee 4, Wis.
(A,B:a,b,e) American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
(A) **Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill.** (*p. 49*)
(B:a,b,c,d) Cutler-Hammer, Inc., 315 N. 12 St., Milwaukee 1, Wis.
(A:a;B:a,b,c) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
(B:a,b,d) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(B:a,b,c) Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
(A,B:a,b,c,e) Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
(B:a,b,c,d,e) Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
(A,B:a,b,c,d,e) Penn Elec. Switch Co., Goshen, Ind. (*p. 69*)
(B:a,b,c,d) Ranco Inc., 601 W. 5th Ave., Columbus 1, O. (*p. 68*)
(A,B:a,d) Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(A,B:a,b) U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
(B:a,b,c,e) White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

CONTROLS, TEMPERATURE ACTUATED (See also THERMOSTATS)

Alco Valve Co., 865 Kingsland Ave., St. Louis 5, Mo. (*p. 99*)
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barber-Colman Co., Rockford, Ill.
Bridgeport Thermostat Div. of Robertshaw-Fulton Controls Co., 1225 Connecticut Ave., Bridgeport 1, Ct.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywel Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
Chas. Engelhard, Inc., 850 Passaic Ave., E. Newark, N.J.
Fischer & Porter Co., Hatboro, Pa.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
Fulton Sylphon Div., Robertshaw-Fulton Controls Co., Knoxville 5, Tenn.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. (*p. 64*)
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
Claud S. Gordon Co. 3000 S. Wallace St., Chicago 16, Ill.
H-B Instrument Co., 2646 Trenton Ave., Philadelphia 25, Pa.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Klipfel Valves, Inc., 1000 Weller Ave., Hamilton, O.
Leeds & Northrup Co., 4970 Stenton Ave., Philadelphia 44, Pa.
Manning Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Penn Elec. Switch Co., Goshen, Ind. (*p. 69*)
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Ranco Inc., 601 W. 5th Ave., Columbus 1, O. (*p. 68*)
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 (*p. 167*)
Spencer Thermostat Co., Unit of Metals & Controls Corp. 34 Forest St., Attleboro, Mass.
Tagliabue Instruments Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon N.Y.
Wheelco Instruments Co., Harrison & Peoria Sts., Chicago 7, Ill.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLLERS, FLOW (See REGULATORS, FLOW)

CONTROLLERS, PRESSURE (See REGULATORS, PRESSURE)

CONTROLLERS, STARTING LOAD

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. (*p. 149*)

CONTROLLERS, TEMPERATURE (See REGULATORS, TEMPERATURE)

CONVERTERS, ELECTRICAL (See RECTIFIERS)

CONVEYORS & COMPONENTS (See also CHAIN; also PULLEYS; also SPROCKETS, etc.)

Ajax Flexible Coupling Co., Inc., Westfield, N.Y. (Vibrating)
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Alvey-Ferguson Co., Oakley Sta., Cin'ti., O.
American Monorail Co., 13107 Athens Ave., Cleveland 7, O.

Baldwin Duckworth Div., Chain Belt Co., 360 Plainfield St., Springfield, Mass.
C. O. Bartlett & Snow Co., 6200 Harvard Ave., Cleveland 5, O.
Berlin Chapman Co., Berlin, Wis.
Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.
Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
E. W. Buschman, Inc., 4400 Clifton Ave., Cin'ti. 32, O.
Cambridge Wire Cloth Co., Cambridge, Md.
Chain-Belt Co., 1600 W. Bruce St., Milwaukee 4, Wis.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Cleveland Crane & Engrg. Co., Wickliffe, O. (Overhead)
Conveyor Co., 3260 E. Slauson Ave., Los Angeles 58, Cal.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp., Subsidiary, P.O. Box 260, Waukegan 1, Ill.
R. & J. Dick Co., Inc., 24 Sade St., Passaic, N.J.
Dodge Mfg. Co., 505 S. Union St., Mishawaka, Ind.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Gifford-Wood Co., Hill St., Hudson, N.Y
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Island Equip. Corp., 101 Park Ave., N.Y.C. 17
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Lamson Corp., Syracuse 1, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Samuel Olson Mfg. Co., Inc. 2418 Bloomingdale Rd., Chicago 47, Ill.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Quaker Rubber Corp., Tacony & Comly Sts., Phila. 24, Pa. (Belting only)
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Read Machinery Div., Standard Stoker Co., Inc., York, Pa.
Robins Conveyor Div., Hewitt-Robins, Inc., 270 Passaic Ave., Passaic, N.J.
Sandvik Steel, Inc., Conveyor Div., 111 8th Ave., N.Y.C. 11
Shaw-Box Crane & Hoist Div., Manning, Maxwell & Moore, Inc., Muskegon, Mich.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill. (Guards)
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

COOLANT COOLERS & COOLANT COOLING SYSTEMS (See also OIL COOLERS)

Acme Industries, Mechanic & Ganson St., Jackson, Mich. *(p. 211)*
Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. *(p. 19)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa. *(p. 57)*
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Read Machinery Div., Standard Stoker Co., Inc., York, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2002 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Taco Heaters, Inc., 137 South St., Providence 3, R.I.

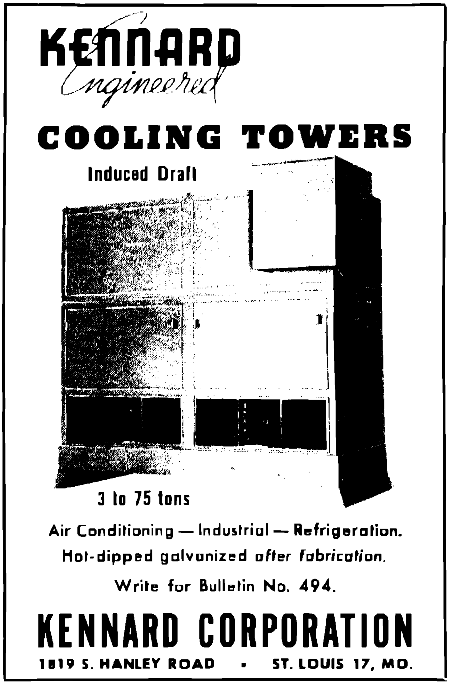
Temprite Products Corp., E. Maple Rd. Birmingham, Mich. *(p. 89)*
Vitter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. *(p. 93)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
B. S. Williams Co., Inc., 6 North St., Mount Vernon 1, N.Y.
Young Radiator Co., Racine, Wis. *(p. 77)*

COOLERS (See particular type)

COOLING TOWERS & PONDS

Acme Equip. Co., 205 E. Broadway, Muskogee, Okla.
Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
Air & Refrigeration Corp., 475 5th Av., New York 17, N.Y.
Airtemp. Div., Chrysler Corp., 1600 Webster, Dayton 1, O. *(p. 19)*
American Cooling Tower Co., 2710 McGee St., Kansas City 8, Mo.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Baltimore Aircoil Co., Inc., 2519 Pennsylvania Ave., Baltimore 17, Md. *(p. 95)*
Belcher-Young, 500 S. Packwood, Tampa, Fla.
Binks Mfg. Co., 3124-32 Carroll Ave., Chicago 16, Ill. *(p. 73)*
Continental Air Filters, Inc., 2520 Helm St., Louisville, Ky.
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa. *(p. 76)*
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 74)*

(Continued)



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COOLING TOWERS
Induced Draft
3 to 75 tons
Air Conditioning — Industrial — Refrigeration.
Hot-dipped galvanized *after fabrication.*
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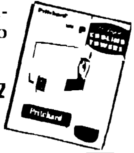
1819 S. HANLEY ROAD • ST. LOUIS 17, MO.

Meet Water Conservation Requirements
Assure Longer Life—Trouble Free Operation
Specify PRITCHARD Cooling Towers

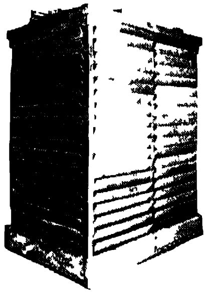
SERIES "P" COOLING TOWERS

New, better-than-ever Pritchard Series "P" Packaged Cooling Towers are manufactured of highest quality materials . . . shipped either "knocked-down" or shop assembled. These *low cost* Induced Draft Towers operate efficiently, economically on air conditioning and refrigeration installations. Refrigeration capacities: 6 to 20 tons.

Write for FREE Bulletin No. 5.6.082



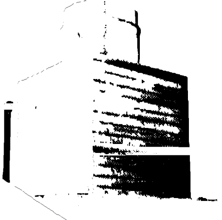
SERIES "SRF" COOLING TOWERS



Pritchard Series "SRF" Cooling Towers are shipped complete with basin . . . Full instructions furnished for quick field assembly. Constructed of finest California Redwood, "SRF" Towers are especially suited to air conditioning installations. *Low* in cost—*high* in operating efficiency. Refrigeration capacities: 3 to 24 tons.

Write for FREE Bulletin No. 5.9.081

SERIES "G" COOLING TOWERS



Pritchard Series "G" Cooling Towers available in either Redwood or steel construction—prefabricated for fast, easy field assembly. Oversize steel pipe water distribution system of special design eliminates nozzles . . . results in unusually low pumping heads. Refrigeration capacities: 20 to 100 tons.

Write for FREE Bulletin No. 5.4.080

Producer
of
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Specialized Process
EQUIPMENT

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J.F. Pritchard & Co.

Dept. No. 62 908 Grand Ave., Kansas City 6, Mo.

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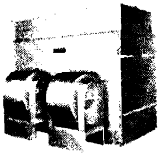
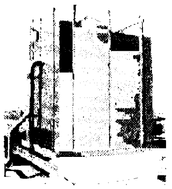
Binks COOLING TOWERS and SPRAY NOZZLES

COOLING TOWERS



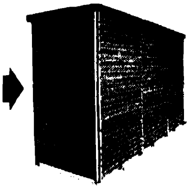
Type 2-B Series—Dry Fan. Recommended wherever quiet operation is essential. Shipped K.D. with easy-to-follow erection instructions. Available in double, squirrel cage blower, 32 to 60 tons (Bulletin 47); or single blower, 3 to 30 tons (Bulletin 46).

Type 2-D Series—Horizontal Induced Draft. Used principally in air conditioning systems or where tower must fit into restricted height. Available in single fan unit, 3 to 90 gpm (ask for Bulletin 33); or twin fan, 150 to 300 gpm (Bulletin 34).



Type K Series—Steel Cased, Induced Draft. Towers in this series are of the spray filled or deck filled type—made in 20 standard sizes. Larger models engineered to specification. V-belt or reduction gear drive for air propulsion. Bulletin 36 describes spray-filled; Bulletin 37, deck-filled.

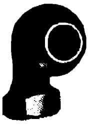
Type S Series—Natural Draft. Ideal for locations where air movement is not restricted. Unit construction simplies erection. Available in capacities from 15 to 125 gpm (ask for Bulletin 30); from 150 to 600 gpm (Bulletin 31), and from 600 to 1200 gpm (Bulletin 32).



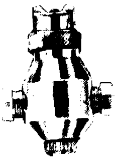
Type K Series—Induced Draft with masonry walls. Highly practical where the tower must fit into the architectural lines of a building. Bulletin 38 contains much practical help on masonry tower design for architects and engineers.

SPRAY NOZZLES

Left: The side-inlet whirl principle of Rotojet nozzles gives exceptionally fine fluid break-up.



Left: Binks Rotojet nozzles are available in small, medium and large capacities. Ask for Bulletins 10, 11 and 12.



Left: Binks pneumatic atomizing nozzles help provide close humidity control. Ask for Bulletin 16.

There's a Bink's Tower and Nozzle for every cooling job.

Binks

BINKS MANUFACTURING COMPANY
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Refrigerant Tables, Charts and Characteristics now on sale in separate book form

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Contains tables and charts on air; ammonia; butane; carbon dioxide; ethane; ethylene; Freon; isobutane; methane; methyl chloride; nitrous oxide; propane; propylene; sulfur dioxide; and others. Also a chapter on refrigerant characteristics.

●

Reprinted from the 1949 Basic Volume of the ASRE Data Books, this new book can be obtained in handy size for $1.50 (paper covered) or $2.00 bound in cloth, from

American Society of
Refrigerating Engineers
40 West 40 Street, New York 18, N.Y.

COOLING TOWERS (Continued)

Kennard Corp., 1819 S. Handley Rd., St. Louis 17, Mo. *(p. 71)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Kramer Trenton Co., Olden & Brenning Aves., Trenton 5, N.J. *(p. 195)*
Larkin Coils, 519 Memorial Drive, S.E., Atlanta 1, Ga. *(p. 41)*
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
Lilie-Hoffmann Cooling Towers, Inc., 4239 Duncan Ave. St. Louis 10, Mo.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 75)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa.
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nevinger Mfg. Co., Greenville, Ill.
New England Cooling Tower Co., 89 Broad St., Boston 10, Mass.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Phillips Cooling Tower Co., Inc., 220 Dupont St., Brooklyn 22, N.Y.
J. F. Pritchard & Co., 908 Grand Ave., Kansas City 6, Mo. *(p. 72)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Appliances, Inc., 923 W. Lake St., Chicago, Ill.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles 22, Cal. *(p. 196)*
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.

(Continued)

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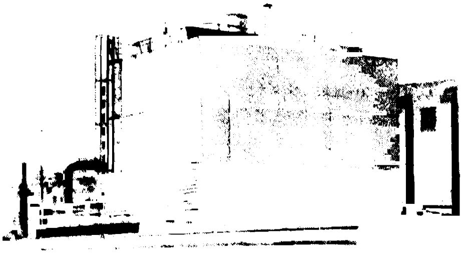

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We introduced and patented Slip Fit construction in our Towers over Fifteen years ago eliminating the use of nails in Harry Framing, Eliminators, and Decks. We also use Slip Fit joints in our standard Transite and Metal casing Mechanical Draft Towers.

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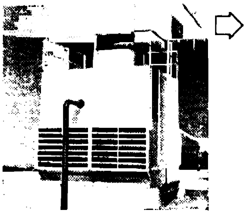
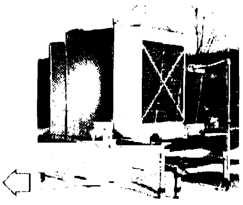
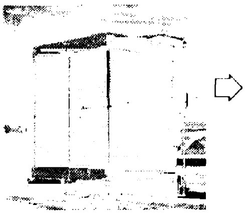
Water Cooling Towers and DriCoolers of All Types and Capacities. Spray Nozzles



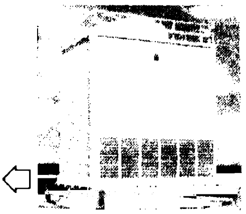
MARLEY Vairflo . . . Sets a new "extra quality" standard for air conditioning and refrigeration cooling towers at no extra cost. Available in choice of steel, wood or asbestos board casing. Has many features formerly found only on heavy duty industrial cooling towers. 25–750 tons of refrigeration. Write for Bulletin V-51.



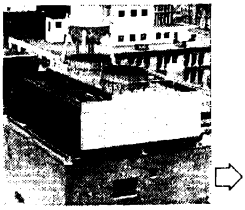
MARLEY Aquatower . . . For "packaged" water cooling, the steel Marley Aquatower packs solid performance for any job, indoors or outdoors. Shipped from stock completely assembled, ready to go, or sub-assembled to eliminate hoisting. Range is from 3 to 60 tons refrigeration; nine sizes available. Write for Bulletin AQ-51.



MARLEY Natural Draft . . . made of heart quality redwood and available in two models. Economical to erect and operate, sturdy and long lasting. Assures top performance. *Series* 100 . . . used primarily for refrigeration and air conditioning small buildings, theaters, locker plants ranging up to 35 tons of refrigeration. *Series* 200 . . . is built in standardized units for larger capacities and is also available with atmospheric sections for indirect cooling of jacket water, oils and gases. Write for Bulletins 100-51 or 200-51.



MARLEY Conventional . . . Marley conventional COUNTER-FLOW towers assure peak performance and operating economy. Built to high structural standards, they are equipped with Marley mechanical units and other exclusive Marley features. Marley Conventional towers meet every water cooling application and are also adaptable to indirect cooling with atmospheric sections. Available in redwood, steel or asbestos board casing. Medium to large capacity. Write for Bulletin C-51.



MARLEY Double-Flow . . . Has these advanced features in cooling tower design: Horizontal air-flow through full height louver walls, Open distribution system, Low pumping head, Minimum draft loss, Double service from each fan unit, Patented, nail-less filling which retains correct alignment without fasteners, Marley designed and built mechanical equipment, Utmost flexibility, Safe operation and low maintenance cost. Available in redwood or asbestos board casing. Double-flows are used wherever large gallonage of water must be cooled economically and efficiently. Write for Bulletin DF-51.

MARLEY Nozzles . . . Marley patented, bronze Spray Nozzles achieve the fundamental requirement of every spray installation . . . a maximum breakup at lowest pressure . . . without moving parts. Three styles, one piece, two piece, humidifying. Write for Bulletin SN3-51.

COOLING TOWERS (Continued)

Snyder Equipment Co., 4115 Edwards Rd., Cincinnati 9, O.
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
Spraying Systems Co., 3201 Randolph St., Bellwood, Chicago, Ill.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Trane Co., La Crosse, Wis. *(p. 118)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
United Conditioning Corp., 74 Varick St., N.Y.C. 13
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd, Minneapolis 14, Minn. *(p. 87)*
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Water Cooling Equip. Corp., Afton Sta., St Louis 23, Mo.
C. H. Wheeler Mfg. Co., 1741 Sedgley Ave., Phila. 32, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.
York Corp., York, Pa. *(p. 52)*

COPPER (See particular mill forms, i.e., BAR, SHEET, etc.)

COPPERSMITHING

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Jos. Kopperman & Sons, 312 New St., Phila. 6, Pa.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.

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CORK (See INSULATION)

CORROSION INHIBITORS (See WATER TREATING)

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Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Lamson & Sessions Co., 1971 W. 8th St., Cleveland 2, O.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

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Boston Gear Works, Quincy 71, Mass.
Browning Mfg. Co., Inc., Maysville, Ky.
Centric Clutch Co., Woodbridge, N.J.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
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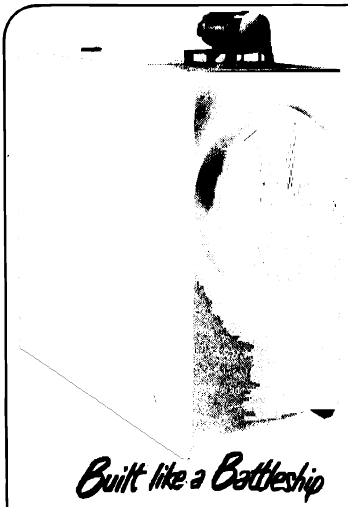
Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Gustin-Bacon Mfg. Co., Kansas City 7, Mo.
Hamilton Kent Mfg. Co., Kent. O. (Flexible)
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 34 Pa.
Victaulic Co. of America, P.O. Box 509, Elizabeth, N.J



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Halstead & Mitchell COOLING TOWERS

Here's 20 years of foolproof performance in an economical and lastworthy cooling tower. Halstead & Mitchell's processed wetted deck surface is chemically treated with special Koppers wood—guaranteed 20 years against rotting and proofed against fungi growth. Gravity-type distributing pan eliminates extra pumping head required on spray-type towers, and cuts down windage losses due to atomizing of water. And, it's easily cleaned —accessible through inlet air side of tower.

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Housing of 10 gauge (1/8") sheet steel, with 3 coats of Bitumastic lining, and 4 coats of outside special weatherproof paint. Electrically welded cabinet, quiet-operating high-pressure, 8-bladed STAINLESS STEEL Fan.

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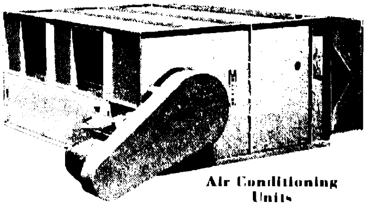
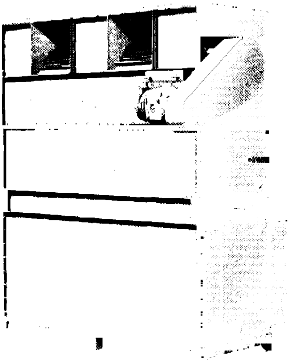
YOUNG AIR CONDITIONING UNITS are available in eight horizontal and eight vertical styles, in capacities from 400 to 16,625 CFM. Designed to fit every size and type installation for heating, cooling, humidifying, dehumidifying and filtering.

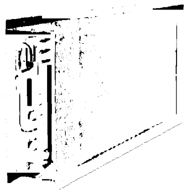


Air Conditioning Units (Horizontal Type)



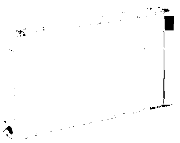
Air Conditioning Units (Vertical Type)

YOUNG DIRECT EXPANSION EVAPORATORS are designed for mechanical refrigerating systems that use Freon or methyl chloride as the cooling agent. Available in a variety of sizes, these units incorporate the new Young liquid distributor.



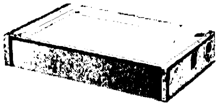
Evaporator Coil

YOUNG COMMERCIAL HEAT TRANSFER UNITS provide the perfect heat transfer surface for application with heating and air conditioning installations where a compact and efficient coil is required.



Commercial Heat Transfer Unit

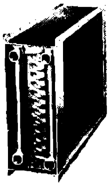
YOUNG BLAST UNITS and BOOSTER UNITS may be used for various applications in connection with forced air heating and conditioning systems. These units are completely encased heat transfer surfaces and are made in a wide range of sizes and capacities.



Blast Unit

YOUNG WATER COILS are designed for use with cold or hot water. Can be installed for cooling alone or as a part of equipment handling complete or partial air conditioning. Type "K" coils have removable headers to simplify cleaning. Type "W" coils have continuous tubes for multi-pass circulation. Both types are available in a wide choice of sizes and capacities.

Type "K" Water Coil



Type "W" Water Coil

YOUNG HEAT TRANSFER PRODUCTS

T. M. REG. U. S. PAT. OFF.

YOUNG RADIATOR CO.
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Paxton-Mitchell Co., 2614 Martha St., Omaha 5, Neb.
Sperry Products, Inc., Danbury, Ct.

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CRADLES, MOTOR (See BASES)

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Cleveland Crane & Engrg. Co., Wickliffe, O.
Euclid Crane & Hoist Co., 136 S. Chardon Rd., Euclid 17, O.
Frick Co., Waynesboro, Pa. (*p. 51*)
Gifford-Wood Co., Hudson, N.Y.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Reading Crane & Hoist Corp., 2100 Adams St., Reading, Pa.
Reco Products Div., Refrigeration Engrg. Corp,. 2020 Naudain St., Phila. 46, Pa.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Robbins & Myers, Inc., 1345 Lagonda Ave., Springfield O.
Shaw-Box Crane & Hoist Div., Manning, Maxwell & Moore, Inc., Muskegon, Mich.
Shepard Niles Crane & Hoist Corp., Schuyler Ave., Montour Falls, N.Y. (Ice)
Union Mfg. Co., New Britain, Ct.

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Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Modern Machine Wks., Inc., 5355 S. Kirkwood Ave., Cudahy, Wis. (*p. 172*)
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
J. H. Williams & Co., 400 Vulcan St., Buffalo 7, N.Y.

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CUPS, GREASE & OIL (See GREASE CUPS; also LUBRICATORS)

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Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Miller Motor Co., 4027 N. Kedzie Ave., Chicago 18, Ill.
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. (*p. 79*)
Scaife Co., Oakmont, Pa.
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.

DAIRY REFRIGERATORS (See DISPLAY CASES)

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Air Conditioning Products Co., 2340 W. Lafayette Blvd., Detroit 16, Mich.
Air Control Products, Inc., Coopersville, Mich.
Aireron Industries, 2536-14th St., Detroit 16, Mich.
American Warming & Ventilating Co., 1017 Summit St., Toledo 4, O.
Anemostat Corp. of America, 16 E. 39th St., N.Y.C. 16 (*p. 14*)
Bahnson Co., 1001 S. Marshall St., Winston-Salem, N.C.
Barber-Colman Co., Rockford, Ill.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Corbman Bros., 315 N. 7th St., Phila. 6, Pa.
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Excelsior Steel Furnace Co., 118 S. Clinton St., Chicago 6, Ill.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. (*p. 10*)
Ludlow Valve Mfg. Co., Inc., Foot of Adams St., Troy, N.Y.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Minneapolis-Honeywell Regulator Co., 2933 4th Ave. S., Minneapolis 8, Minn. (*p. 67*)
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
H. H. Robertson Co., 2400 Farmers' Bank Bldg., Pittsburgh 22, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
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Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O.

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Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Conoflow Corp., 2100 Arch St., Phila. 3, Pa.
Excelsior Steel Furnace Co., 118 S. Clinton St., Chicago 6, Ill.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Johnson Service Co., 507 E. Michigan Ave., Milwaukee 22, Wis.
Mason-Nielan Regulator Co., 1190 Adams St., Boston 24, Mass.
Minneapolis-Honeywell Regulator Co., 2933 4th Ave. S., Minneapolis 8, Minn. (*p. 67*)
H. H. Robertson Co., 2400 Farmers' Bank Bldg., Pittsburgh 22, Pa.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Schade Valve Mfg. Co., 2527 Bodine St., Phila. 33, Pa.
Tagliabue Instruments, Div. Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Tuttle & Bailey, Inc., New Britain, Ct.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O.

DECORATIONS, METAL (See NAMEPLATES)

DEFROSTERS, AUTOMATIC (See CONTROLS, DEFROSTING CYCLE)

DEHUMIDIFYING AGENTS (See DEHYDRANTS)

DEHUMIDIFYING SYSTEMS & UNITS

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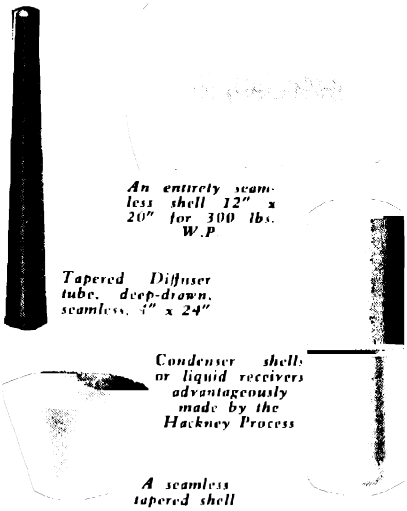
Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. (*p. 19*)
American Coils Co., 25 Lexington St., Newark 5, N.J.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Blissfield Mfg. Co., 626 Depot St., Blissfield, Mich.
Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.

(Continued)

DEEP DRAWN SHAPES AND SHELLS AND CONTAINERS

Hackney Seamless Drawn Shells are ideal for use as condenser shells and refrigerant gas receivers. Efficient performance is due to their smooth surface, elimination of scale, neat appearance, comparative light weight and economy. Pioneers in the cold drawing of seamless containers from metal plates, Pressed Steel Tank Company has had more than 45 years' experience in designing, engineering and manufacturing containers for gases, liquids and solids. Our engineers will gladly work with you in developing containers or special shapes to your individual needs. For full details, write us today.



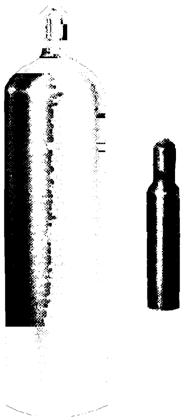
An entirely seamless shell 12" x 20" for 300 lbs. W.P.

Tapered Diffuser tube, deep-drawn, seamless, 4" x 24"

Condenser shells or liquid receivers advantageously made by the Hackney Process

A seamless tapered shell

REFRIGERANT COMPRESSED GAS CYLINDERS



Hackney light tare weight cylinders are widely used for anhydrous ammonia, freon, sulphur dioxide, methyl chloride, iso-butane, propane, etc.—in capacities from 25 pounds up. They have been designed with the users' requirements in mind—constructed to give full protection to product, ease of handling, utmost resistance to transportation and handling abuse and lowest transportation costs.

Hackney refrigeration cylinders are of two-piece construction or are entirely seamless—formed by a series of cupping cold drawing operations—and subjected to special laboratory controlled heat-treatment, after complete frabrication, to further improve physical qualities. Hackney cold drawing process provides uniform sidewall thickness and eliminates excess material.

All Hackney ICC Cylinders are manufactured to specifications that more than comply with the minimum requirements set up under Interstate Commerce Commission Regulations. Write Pressed Steel Tank Company for full information.

Hackney

Pressed Steel Tank Company

Manufacturers of Hackney Products

1483 S. 66th St., Milwaukee 14
1389 Vanderbilt Concourse Bldg., New York 17
231 Hanna Bldg., Cleveland 15

936 W. Peachtree St., N.W., Room 124, Atlanta 3
208 S. La Salle St., Room 776, Chicago 4
566 Roosevelt Bldg., Los Angeles 14

DEHUMIDIFYING SYSTEMS & UNITS (Continued)

Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Dry Air Products Corp., 734 Jackson Place, Washington 6, D.C.
Drying System, Inc., 1810½ Foster Ave., Chicago 40, Ill
Drynamic Div., Cargocaire Engrg. Corp., 15 Park Row, N.Y.C. 7
Dryomatic Corp. of America, 1600 Union Ave., Baltimore 11, Md.
Ebco Mfg. Co., 401 W. Town St., Columbus 8, O.
Frigidaire Div., Gen'l Motors Corp., Dayton 1, O. *(p. 17)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Electric Co., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
W. A. Hammond Drierite Co., 120 Dayton Ave., Xenia, O. *(p. 80)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Mitchell Mfg. Co., 2525 Clybourn Ave., Chicago 14, Ill.
D. J. Murray Mfg. Co., 1024-3rd St., Wausau, Wis.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
Pittsburgh Lectrodryer Corp., P.O. Box 1766, Pittsburgh 30, Pa.
Refrigeration Corp. of America, Div. of Lonergan Mfg. Co., Inc., Albion, Mich.
Remington Corp., Willey St., Auburn, N.Y.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 23)*
Surface Combustion Corp., 2375 Dorr St., Toledo 1, O.
Trane Co., La Crosse, Wis. *(p. 118)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd, Minneapolis 14, Minn. *(p. 37)*
York Corp., York, Pa. *(p. 52)*

DEHYDRANTS

Aluminum Co. of America, Pittsburgh, Pa.
Davison Chemical Corp., Baltimore 3, Md. (Silica Gel) *(p. 81)*
Floridin Co., Warren, Pa.
W. A. Hammond Drierite Co., 120 Dayton Ave., Xenia, O. *(p. 80)*
Highside Chemicals Co., 10 Colfax Ave., Clifton, N.J.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*
Michigan Alkali Div., Wyandotte Chemicals Corp., Wyandotte, Mich.
Socony Vacuum Oil Co., 26 Broadway, N.Y.C. 4
Surface Combustion Corp., 2375 Dorr St., Toledo 1, O.
Tamms Industries, Inc., 228 N. La Salle St., Chicago 1, Ill.

DEHYDRATORS (See DRIERS)

DELICATESSEN REFRIGERATORS (See DISPLAY CASES)

DEODORIZERS (See AIR PURIFICATION EQUIPMENT)

DESICCANTS (See DEHYDRANTS)

DESICCATORS (See DRIERS)

DIAPHRAGMS

Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Black, Sivalls & Bryson, Kansas City 3, Mo.
Darcoid Co., Inc., 145-6th Ave., N.Y.C. 13
Goshen Rubber Co., Box 517, Goshen, Ind.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.

(Continued)

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DIAPHRAGMS (Continued)

New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

DIALS, GAUGE, etc. (See NAMEPLATES)

DICHROMATE (See WATER TREATING MATERIALS)

DIE CASTINGS (See CASTINGS, DIE)

DIFFUSERS, AIR (See AIR DIFFUSERS)

DISCS, COMPRESSOR VALVE (See VALVE DISCS, FLAPPERS, REEDS, etc.)

DISCS, METAL, COMPOSITION, etc.

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Black, Sivalls & Bryson, Kansas City 3, Mo.
Climax Controls, Div. of Black, Sivalls & Bryson, Inc., 15 N. Cincinnati, Tulsa, Okla.
Delgro Mfg. Co., Inc., Boonton, N.J.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind.
Jenkins Bros., 80 White St., N.Y.C. 13

DISHES, REFRIGERATOR

Corning Glass Wks., Corning, N.Y.
Sneath Glass Co., Hartford City, Ind.
Amos Thompson Corp., Edinburgh, Ind.

DISPENSING MACHINES, REFRIGERATED (See VENDING MACHINES)

DISPLAY CASES, FROZEN FOOD

Anheuser-Busch, Inc., 9th & Wyoming, St. Louis, Mo.
Brewer-Titchener Corp., Binghamton, N.Y.
Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
Ed Friedrich, Inc., 1117 E. Commerce St., San Antonio 3, Tex.
Harder Refrigerator Corp., Div. of Tyler Fixture Corp., Cobleskill, N.Y.
John Herrel & Sons, Inc., 244 Lear St., Columbus 6, O.
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
Jordan Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, Pa.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Koch Refrigerators, 600 E. 14th Ave., N. Kansas City 16, Mo.
Minneapolis Show Case & Fixture Co., New Richmond, Wis.
Morton Show Cases, Inc., Washington Court House, O.
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
Refrigeration Corp. of America, Div. of Lonergan Mfg. Co., Inc., Albion, Mich.

(Continued)

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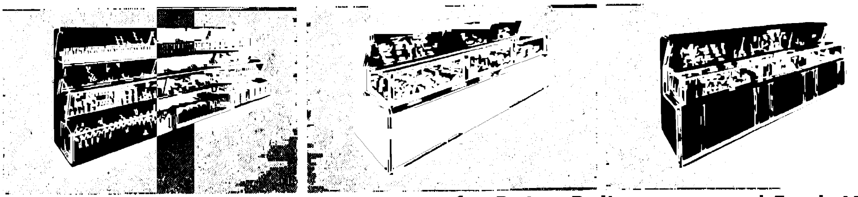

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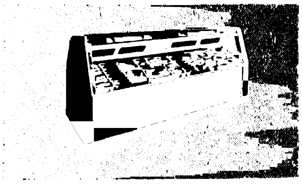
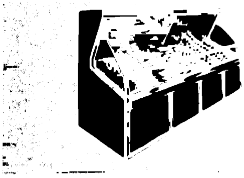
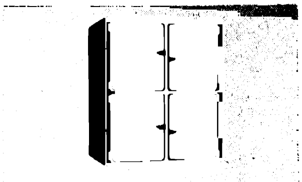


Continuous
Top Service
Meat Case

Self-Service Open
Display Refrigerator
for Produce





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In Many Sizes
Solid Doors or Glass

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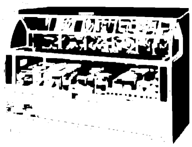
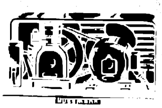
Refrigerator for
Frozen Foods





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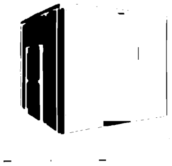

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Hussmann Refrigerator Co., Limited, Brantford, Ont., Hussmann Canadian Subsidiary

DISPLAY CASES, FROZEN FOOD (Continued)

Schaeffer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
C. Schmidt Co., 1712 John St., Cincinnati 14, O.
Sherer-Gillett Co., Marshall, Mich.
Super-cold Cold Corp., 1020 E. 59th St., Los Angeles, Cal.
Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 82)*
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.

DISPLAY CASES, REFRIGERATED

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) American Refrigeration Corp., 1025 E. Excelsior Ave., Hopkins, Minn.
(A,B) Anheuser-Busch, Inc., 9th & Wyoming St. Louis Mo.
(A,B) Bally Case & Cooler Co., Bally, Pa.
Brewer-Titchener Corp., Binghamton, N.Y.
(A) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B) Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
(A,B,C) Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B,C) Fleetwood-Airflow, Inc., 421 N. Penna. Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
(A,B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 3, Tex.
(A,B,C) Gem Refrigerator Co., 165 W. Wyoming Ave., Phila. 40, Pa.
(A,B) John Herzel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(A) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
(A,B,C) Jordon Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, P.
Koch Refrigerators, 600 E. 14th Ave., N., Kansas City 16, Mo.
(A) Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
(A) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
(B) Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
(A,B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
(A,B,C) Minneapolis Show Case & Fixture Co., New Richmond, Wis.
(A,B) Morton Show Cases, Inc., Washington Courthouse O.
(A,B) John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
(A,B) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
C. L. Percival Refrigerator Co., 1805 E. 4th St., Boone, Ia.
(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(A) W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
(A) Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Southern Fixture Mfg. Co., P.O. Box 245, Greensboro, N.C.
(B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 82)*
(A) Victor Products Corp., Hagerstown, Md.
(A,B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B,C) Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.
(A,B,C) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.

DISPLAY CASE DOORS & FRAMES

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Evans Mfg. Corp., 460 S. 10th St., Mt. Vernon, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
Jordon Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, Pa.
Luzerne Rubber Co., Trenton 9, N.J.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
Amos Thompson Corp., Edinburgh, Ind.

DISTRIBUTORS, CONDENSER WATER (See CONDENSER WATER DISTRIBUTORS)

DISTRIBUTORS, REFRIGERANT

Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.

DOOR & FRAME ASSEMBLIES (See DISPLAY CASE DOORS or COLD STORAGE DOORS)

DOORS, COLD STORAGE (See COLD STORAGE DOORS)

DOORS, DISPLAY CASE (See DISPLAY CASE DOORS)

DOORS, EVAPORATOR

Aluminum Goods Mfg. Co., Manitowoc, Wis.
Anderson and Sons, Inc., N. Elm St., Westfield, Mass.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Evans Mfg. Corp., 460 S. 10th St., Mt. Vernon, N.Y.
Gen'l Industries Co., Olive & Taylor Sts., Elyria, O.
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Reynolds Metals Co., 2000 S. 9th St., Louisville, Ky.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Sneath Glass Co., Hartford City, Ind.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Amos Thompson Corp., Edinburgh, Ind.

DOORS, FREEZER (See COLD STORAGE DOORS)

DOOR GASKET

Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Armstrong Cork Co., Lancaster, Pa.

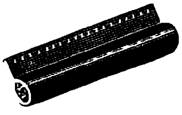
(Continued)

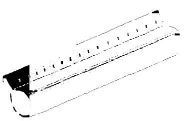
DOOR GASKETS

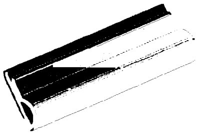
FOR REFRIGERATORS & COOLING ROOMS

For nearly a quarter of a century JARROW PRODUCTS has specialized in designing refrigerator door gaskets to meet the most exacting standards of original equipment manufacturers. Embodying matchless quality with precision workmanship, Jarrow Door Gaskets offer the ultimate in efficiency and endurance.

A

B


C


D


A Water proofed for low temperature applications Is web covered choice of cotton or sponge rubber cushion

B Has sponge rubber cushion with rubberized fabric covering Also furnished with grease proof cover

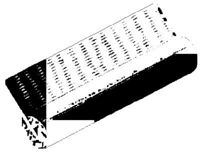
C Cotton cushion with rubberized fabric covering in black or white Grease proof in black only

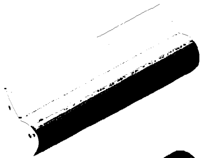
D Extruded all rubber Resilient odorless and non staining All shapes and sizes to fit any refrigerator or to blueprint

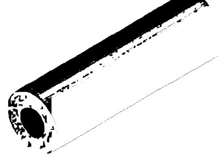
E Innerseal gasket with sponge rubber cushion and rubber covered woven wire flange Withstands sub zero temperatures

F Molded Sponge Rubber gasket Has grease proof Neoprene jacket

G Sponge rubber tubing for insulating cold compressor lines

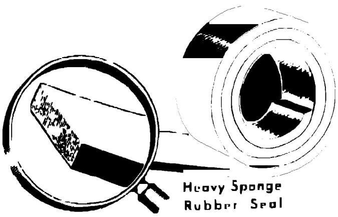
E


F


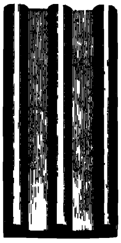
G


HEAVY DUTY SEALING STRIP

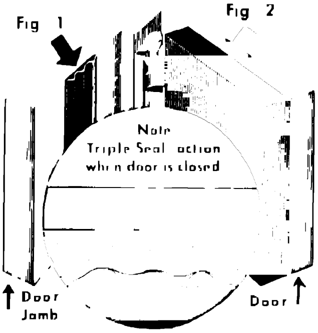
For Heavy Freezer Doors, Walk-In Coolers, Refrigerated Trucks, Etc.



Heavy Sponge Rubber Seal



Hard Rubber Triple Seal



A positive, air tight seal is assured with this Heavy Duty Sponge Rubber Strip compressed under door pressure against a ribbed faced hard rubber triple seal

Sponge rubber seal is supplied with Neoprene or Molded Skin coating **Fig 1** shows installation of hard rubber seal on door jamb **Fig 2** illustrates sponge rubber seal cemented to door

---

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Manufacturers:

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DOOR GASKET (Continued)

Ball Bros. Co., Muncie, Ind.
Brasco Mfg. Co., Harvey, Ill.
Bridgeport Fabrics Co., Bridgeport 1, Ct.
Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
A. W. Chesterton Co., 4 Ashland St., Everett 49, Mass.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
W. J. Dennis & Co., 4444 Irving Park Rd., Chicago, Ill.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 171)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Jamison Cold Storage Door Co., Hagerstown, Md.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 85)*
Johns, Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Quaker Rubber Corp., Tacony & Comly Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Rubatex Div., Great American Industries, Bedford, Va. *(p. 130)*
Sponge Rubber Products Co., 1600 Derby Place, Shelton, Ct.
Standard-Keil Hardware Mfg. Co., Inc., 639 B'way, N.Y.C. 12
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

DOUBLE DUTY REFRIGERATORS (See DISPLAY CASES)

DOUGH RETARDERS

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(A,B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B,C) Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Freezer Box Div., Ananpolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 3, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B,C) Gem Refrigerator Co., 165 W. Wyoming Ave., Phila. 40, Pa.
(A,B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 165)*
(A,B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
(A,B) Jordon Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, Pa.
(A,B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
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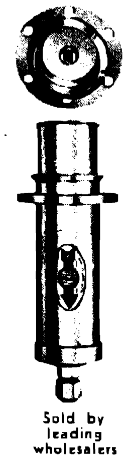

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(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 82)*
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Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 10, N.Y. *(p. 59)*
Fine Products Co., 185 N. Wabansia Ave., Chicago 1, Ill. *(p. 168)*
Henry Valve Co., Melrose Park, Ill. *(p. 88)*
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Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
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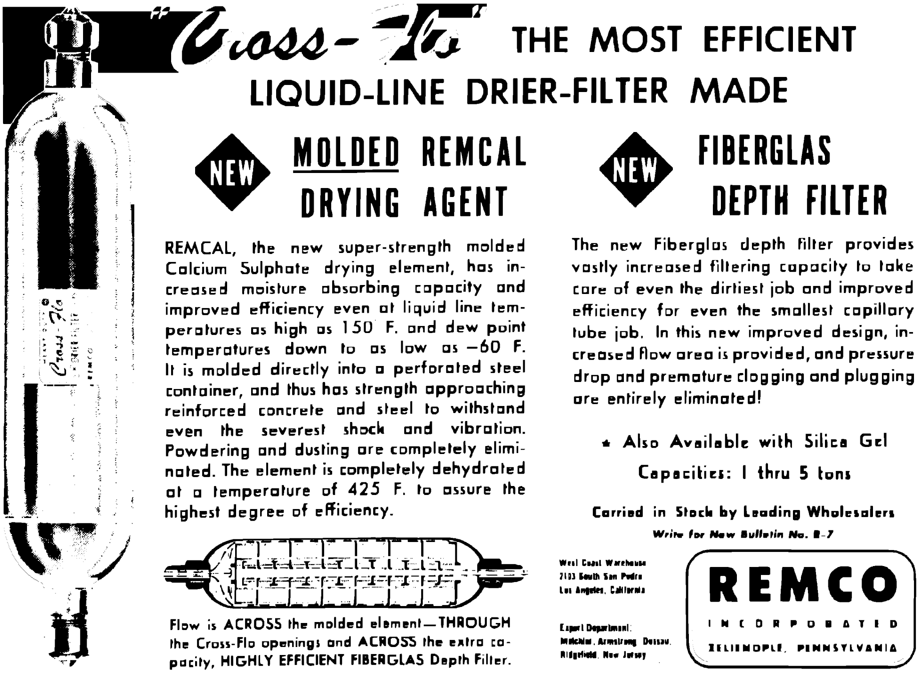
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McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*
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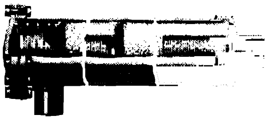

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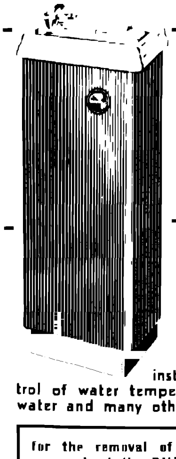

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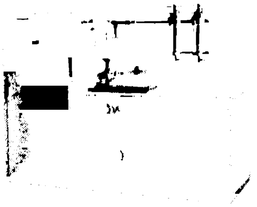

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Westinghouse Elec. Corp., P.O. Box 7318, S. Phila. 1, Pa.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5 N.Y.
C. H. Wheeler Mfg. Col, 1741 Sedgley Ave., Phila. 32, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

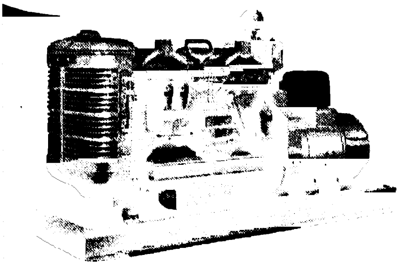
ELECTRIC GENERATING PLANTS, DIESEL OR GASOLINE DRIVEN (See also MOTOR GENERATORS)

Buda Co., 15th & Commercial Ave., Harvey, Ill
Caterpillar Tractor Co., Peoria 8, Ill.
Cummins Engine Company, Inc., Columbus, Ind
Detroit Diesel Engine Div., Gen'l Motors Corp., B400 W. Outer Drive, Detroit 28, Mich. (Standby)
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Hobart Bros. Co., 146 Hobart Square, Troy, O.
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.
Murphy Diesel Co., 5317 W. Burnham St., Milwaukee 14, Wis.
National Supply Co., Engine Div., 1401 Sheridan Ave., Springfield, O.
D. W. Onan & Sons, Inc., University Ave., S.E., at 25th, Minneapolis 14, Minn.
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. *(p. 92)*
Sterling Engine Co., 1260 Niagara St., Buffalo 13, N.Y.
Superior Engine Div., National Supply Co., 1401 Sheridan Ave., Springfield, O.
Universal Motor Co., 430 Universal Drive, Oshkosh, Wis.
U. S. Motors Corp., 559 Nebraska St., Oshkosh, Wis.
Waukesha Motor Co., Waukesha, Wis.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Witte Engine Wks., Div. of Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

ELECTRIC HEATER ELEMENTS (See HEATER ELEMENTS)

ELECTRICAL WIRING HARNESSES (See WIRING DEVICES)

READY-POWER
ENGINE GENERATORS
for Dependable, Low Cost, Independent Electric Power



Powered by Industrial Engines Using Gasoline *Natural Gas* or *Diesel Fuel*

Generate electric power anywhere at low cost. Ideal for stationary, portable or marine use. 2 to 100 KW—AC or DC.

THE READY-POWER CO.
11231 FREUD AVE., DETROIT 14, MICHIGAN.

ELECTRODE COOLING SYSTEMS & COMPONENTS

Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
Universal Welder Corp., 735 Carnegie Ave., Cleveland 15, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

ELECTRODES, WELDING (See WELDING ELECTRODES & RODS)

ELEVATORS, BUCKET (See CONVEYORS)

ELIMINATORS (See particular type, i.e., ODOR, SPRAY, VIBRATION ABSORBERS, etc.)

ELLS (See FITTINGS)

EMBLEMS, METAL (See NAMEPLATES)

ENAMEL (See FINISHES)

ENAMEL, BITUMASTIC (See COMPOUNDS, MASTIC)

ENAMELING, PORCELAIN (See also NAMEPLATES, etc.)

Acromark Co., 5 Morrell St., Elizabeth 4, N.J.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Challenge Stamping & Porcelain Co., Grand Haven 22, Mich.
Chicago Vitreous Enamel Products Co., 1401-47 S. 55th Court, Cicero 50, Ill.
Fox Co., Fox Lane, Cin'ti. 23, O.
Heintz Mfg. Co., Front St. & Olney Ave., Philadelphia 20, Pa.
Ingram-Richardson Co., 32nd St., Beaver Falls, Pa.
Ingram-Richardson Co. of Indiana, Frankfort, Ind.
Mullins Mfg. Co., S. Ellsworth St., Salem, O.
Strong Mfg. Co., Sebring, O.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

ENCLOSURES, WIRE (See WIRE FORMING)

ENGINES, BUTANE, GAS, etc.

Bruce-Macbeth Engine Co., 2111 Center St., N.W., Cleveland 13, O.
Buda Co., 15th & Commercial Ave., Harvey, Ill.
Clark Bros. Co., Inc., Olean, N.Y.
Continental Motors Corp., Muskegon 82, Mich.
Cooper-Bessemer Corp., Mt. Vernon, O.
Cummins Engine Company, Inc., Columbus, Ind.
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.
McCulloch Motors Corp., 6101 W. Century Blvd., Los Angeles 45, Cal.
Sterling Engine Co., 1260 Niagara St., Buffalo 13, N.Y.
Waukesha Motor Co., Waukesha, Wis.
Western Engine Co., Div. of Vernon Tool Co., Ltd., 1101 Meridian Ave., Alhambra, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

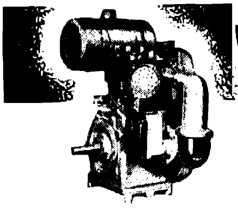
ENGINES, DIESEL

American Locomotive Co., 30 Church St., N.Y.C
Baldwin Locomotive Wks., Phila. 42, Pa.
Bolinders Co., Inc., 33 Rector St., N.Y.C.
Buckeye Machine Co., E. O'Connor Ave., Lima, O.
Buda Co., 15th & Commercial Ave., Harvey, Ill.
Busch-Sulzer Bros. Diesel Engine Co., St. Louis 18, Mo.
Caterpillar Tractor Co., Peoria 8, Ill.
Clark Bros. Co., Inc., Olean, N.Y.
Cleveland Diesel Div., Gen'l. Motors Corp., 2160 W 106th St., Cleveland 11, O.
Continental Motors Corp., Muskegon 82, Mich.
Cooper-Bessemer Corp., Mt. Vernon, O.
Cummins Engine Co., Inc., Columbus, Ind.

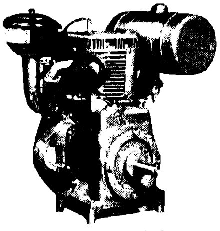
(Continued)

BETTER Engineering Design!
BETTER Versatility! BETTER Economy!

WISCONSIN HEAVY-DUTY *Air-Cooled* Engines

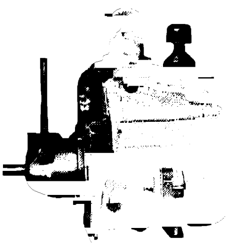


Models ABN and AKN,
Single cylinder, 3 to 6 hp.



Model AEN 7½ hp.
Other Single Cyl. Models
6 to 9 hp.

Models TE and TF,, two-cylinder, 7 to 13 hp.



Models VE-4, VF-4 and VP-4D, V-type 4 cylinder, 15 to 30 hp.

Wisconsin Air-Cooled Engines offer the refrigeration equipment manufacturer and power user the cumulative benefits of many years of exclusive specialization in the manufacture of air-cooled engines, in a complete power range from 3 to 30 H.P.

Every Wisconsin Engine, from the smallest to the largest, is of heavy-duty design and construction, built to the highest quality and service standards — to deliver "Most H. P. Hours" of on-the-job power performance.

Every Wisconsin Engine is equipped with tapered roller bearings at BOTH ends of the accurately balanced, drop-forged crankshaft to take up end- and radial thrusts and provide the greatest protection against bearing failure — also permitting safe mounting of drive pulley, gear or sprocket directly on the extended crankshaft.

Every Wisconsin Engine is extremely compact and relatively light in weight for convenient mounting on a great variety of equipment, without costly re-designing or excess weight handicaps.

Every Wisconsin Engine is provided with rotary type high tension OUTSIDE magneto, sealed against dust and moisture, with impulse coupling for quick, easy starting at any temperature, in any weather, assuring thoroughly dependable ignition.

Every Wisconsin Engine gets a 4-hour test (the last hour under full load) before leaving the factory.

SPECIFICATIONS 2- and 4-CYLINDER MODELS

MODEL	TE	TF	VE-4	VF-4	VP-4 D
Bore	3"	3¼"	3"	3¼"	3½"
Stroke	3¼"	3¼"	3¼"	3¼"	4"
No. of Cylinders	2	2	4	4	4
Displ. Cu. Inches	45.9	53.9	91.9	107.7	154
H. P. and R. P. M. Range	7.2 @ 1400 11.2 @ 2600	8.6 @ 1400 13.2 @ 2600	15 @ 1600 21.5 @ 2400	17.5 @ 1600 25 @ 2400	26.8 @ 1600 31 @ 2200
Net Weight in Lbs. (Standard Engine)	220 (with Side Mount Tank)	220 (with Side Mount Tank)	295	295	410

SPECIFICATIONS, SINGLE CYLINDER MODELS

MODEL	ABN	AKN	AEN	AFH	AGH	AHH
Bore	2½"	2⅞"	3"	3¼"	3½"	3⅝"
Stroke	2¾"	2¾"	3¼"	4"	4"	4"
No. of Cylinders	1	1	1	1	1	1
Displ. Cu. Inches	13.5	17.8	23	33.2	38.5	41.3
H. P. and R. P. M. Range	2.2 @ 1600 4.5 @ 3400	3. @ 1600 5.8 @ 3200	4.4 @ 1600 7.5 @ 3000	6.0 @ 1600 7.2 @ 2200	7.2 @ 1600 8.4 @ 2200	7.7 @ 1600 9.2 @ 2200
Net Weight in Lbs. (Standard Engine)	76	77	110	130	180	180

SPECIAL EQUIPMENT: Variable speed controls, clutch take-off assemblies and reduction assemblies can be furnished on all models. Clutch reduction assemblies and electric starting available for large single cylinder and all two- and four-cylinder models. Special equipment must be specified at time engine is built.

Wisconsin engineers will be glad to work with you in planning your Refrigeration and Conditioner power installations. Let us have a look at your blueprints and specifications and see what we can come up with.

MOST H.P. HOURS

WISCONSIN MOTOR CORPORATION
World's Largest Builders of Heavy-Duty Air-Cooled Engines
MILWAUKEE 46, WISCONSIN

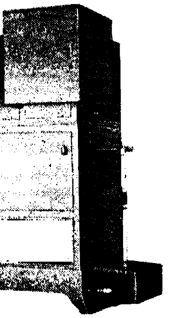
ENGINES, DIESEL (Continued)

Detroit Diesel Engine Div., Gen'l. Motors Corp., 13400 W. Outer Dr., Detroit 28, Mich.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Ingersoll Rand Co., 11 Broadway, N.Y.C. 4
Lima-Hamilton Corp., 545 N. 3rd St., Hamilton, O.
National Supply Co., Engine Div., 1401 Sheridan Ave., Springfield, O.
Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7, Wis.
Palmer Bros. Engine Co., Cos Cob, Ct.
Sterling Engine Co., 1200 Niagara St., Buffalo 13, N.Y.
Superior Engine Div., National Supply Co., 1401 Sheridan Ave., Springfield, O.
United Engine Co.,W. Holmes Rd., Lansing 12, Mich.
Waukesha Motor Co., Waukesha, Wis.
Witte Engine Wks., Div. of Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Wolverine Motor Wks., Inc., 35 Union Ave., Bridgeport 2, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

ENGINES, GASOLINE, KEROSENE

Briggs & Stratton Corp., 2711 N. 13th St., Milwaukee 1, Wis.
Continental Motors Corp., Muskegon 82, Mich.
Fairbanks, Morse & Co., 600 S. Michigan Av., Chicago 5, Ill.
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.
Novo Engine Co., 702 Porter St., Lansing 5, Mich.
D. W. Onan & Sons, Inc., University Ave., S.E. at 25th, Minneapolis 14, Minn.
Salisbury Motor Div., Wayne Mfg. Co., 1201 Lexington Ave., Pomona, Cal.

KENNARD Engineered
EVAPORATIVE CONDENSERS
FREON . . . AMMONIA
3 to 75 tons



Air Conditioning — Industrial — Refrigeration.
Hot-dipped galvanized after fabrication.
Write for Bulletin No. 491.
KENNARD CORPORATION
1819 S. HANLEY ROAD • ST. LOUIS 17, MO.

Sterling Engine Co., 1260 Niagara St., Buffalo 13, N.Y.
United Engine Co., W. Holmes Rd., Lansing 12, Mich.
Waukesha Motor Co., Waukesha, Wis.
Wisconsin Motor Corp., Milwaukee 46, Wis. *(p. 93)*
Witte Engine Wks., Div. of Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

ENGINES, STEAM

Ames Iron Wks., Oswego, N.Y.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O
Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7, Wis.
Skinner Engine Co., Erie, Pa.
Troy Engine & Machine Co., 400 Railroad Ave., Troy, Pa.
United Engine Co., W. Holmes Rd., Lansing 12, Mich.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 35)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

ENGINE JACKET-WATER COOLERS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N. J.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 81)*
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Frick Co., Waynesboro, Pa. *(p. 51)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
Nordberg Mfg. Co., 3073 S. Chase Ave., Milwaukee 7, Wis.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. *(p. 118)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

ENGRAVING (See NAMEPLATES)

EQUALIZER TANKS (See ACCUMULATORS)

ESCUTCHEONS (See NAMEPLATES)

EVAPORATIVE CONDENSERS

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Coils Co., 25 Lexington St., Newark 5, N.J.
American Cooling Tower Co., 2710 McGee St., Kansas City 8, Mo.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Baltimore Aircoil Co., Inc., 2519 Pennsylvania Ave., Baltimore 17, Md. *(p. 95)*
Betz Corp., 445 State St., Hammond, Ind.
Buffalo Forge Co., 217 Mortimer St., Buffalo 1, N.Y. *(p. 98)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y *(p. 25)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal
Frick Co., Waynesboro, Pa. *(p. 51)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 39)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Governaire Corp., 513 N. Blackwelder, Oklahoma City, Okla.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 74)*

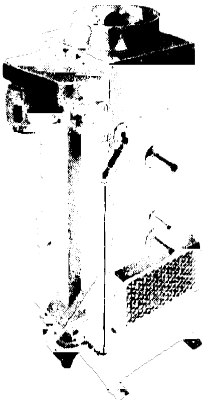
(Continued)

**Fit the condenser to the job . . .
instead of the job to the condenser!**

B.A.C.

EVAPORATIVE CONDENSERS

World's Widest Range—55 Sizes and Styles—3 to 210 TR



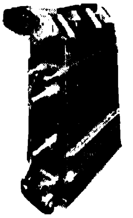
Model "P"

3 to 100 TR

Freon or Ammonia

•

Outdoor Installation
Only



Model "U"

10 to 100 TR

Freon or Ammonia

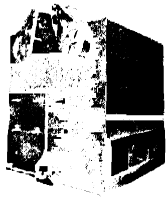
Indoor-Outdoor

Model "UL"

105-210 TR

Freon or Ammonia

Indoor-Outdoor



B.A.C.

**COOLING
TOWERS**

Model "PT"

3 to 100 TR

Outdoor Installation
Only

Model "UT"

7½ to 100 TR

Indoor-Outdoor

For further information, write to:

Baltimore Aircoil Company, Inc.
BALTIMORE, MARYLAND
Specializing in Water-Saving Devices Since 1938

EVAPORATIVE CONDENSERS (Continued)

C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. (*p. 94*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. (*p. 153*)
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
Larkin Coils, 519 S. Memorial Dr., S.E., Atlanta 1, Ga. (*p. 41*)
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. (*p. 200*)
Nevinger Mfg. Co., Greenville, Ill.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
J. F. Pritchard & Co., 908 Grand Ave., Kansas City 6, Mo. (*p. 72*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. (*p. 201*)
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
R. E. Ristow, 2228 S. Atlantic Blvd., Los Angeles 22, Cal.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St.,, Boston 36, Mass. (*p. 23*)
Tenney Engrg., Inc., 26 Ave B., Newark 5, N.J.
Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc.. 794 Union St., Brooklyn 15, N.Y.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 37*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Westinghouse Elec. Corp., P.O. Box 7348, S. Phila. 1, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)
Young Radiator Co., Racine, Wis. (*p. 77*)

EVAPORATIVE COOLERS (See ATMOSPHERIC COOLERS)

EVAPORATORS (See COILS, UNIT COOLERS, etc.)

EVAPORATORS, HOUSEHOLD REFRIGERATOR

Brewer Titchener Corp., Binghamton, N.Y.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. (*p. 152*)
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Houdaille-Hershey Corp., 1900 Foss Park Ave., North Chicago, Ill. (*p. 96*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4 Mich. (*p. 153*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
Rudy Mfg. Co., Dowagiac, Mich.
Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago, 12, Ill.
Tenney Engrg., Inc , 26 Ave B., Newark 5, N.J.

EXCHANGERS, HEAT (See HEAT EXCHANGERS)

EXPANDED METAL (See WIRE CLOTH)

EXPANSION JOINTS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Cleveland Coppersmithing Wks., 5500 Stone Ave., Cleveland 2, O.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Crane Co., 836 S. Michigan Ave., Chicago 5, Ill. (*p. 109*)
Croll-Reynolds Engrg. Co., Inc., 17 John St., N.Y.C.
Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Fulton Sylphon Division, Robertshaw-Fulton Controls Knoxville 4, Tenn.
Garlock Parking Co., 402 E. Main St., Palmyra, N.Y.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Joseph Kopperman & Sons, 312 New St., Phila. 6, Pa.
Reeves Steel & Mfg. Co., 137 E. Iron Ave., Dover, O.
Ric-Wil Co., Union Commerce Bldg., Cleveland 14, O.
Rubatex Div., Great American Industries, Bedford, V.
Walworth Co., 60 E 42nd St., N.Y.C. 17
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

EXPANSION VALVES, AUTOMATIC

(A—Ammonia; B—Methyl Chloride; C—Sulfur; D—Freon)

(A,B,C,D) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. (*p. 99*)
(B,C,D) A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
(A) Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
(A,B,C,D) A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
(B,C,D) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
(B,C,D) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(A) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(A,B,C,D) Hammel-Dahl Co., 243 Richmond St., Providence 3. R.I.
(B,C,D) Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.
(A,B,D) Refrigerating Specialities Co., 728 Sacramento Blvd., Chicago 12, Ill.
(A) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
(A,B,D) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,D) York Corp., York, Pa. (*p. 52*)

EVAPORATORS

Hydrogen Brazed • Internally Clean • Durable Finish • Efficient Circuits • Low Cost • Modern Production Lines • Years of Experience

All of these factors make this organization an outstanding source of supply for production quantities of household refrigerator evaporators and condensers.

HOUDAILLE-HERSHEY CORP.

North Chicago Division, North Chicago, Illinois

Niagara Blower Company

General Sales Office: 405 Lexington Ave., New York 17, N.Y.

Field Engineers in Principal Cities

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Products:

NIAGARA AIR CONDITIONERS— Type "C"—Bulletin #80

NIAGARA CONTROLLED HUMIDITY METHOD— Bulletin #112

NIAGARA NO-FROST METHOD— Bulletins #105, #118, #119

NIAGARA SPRAY COOLERS— Bulletin #110

NIAGARA DUAL COOLERS— Bulletin #70

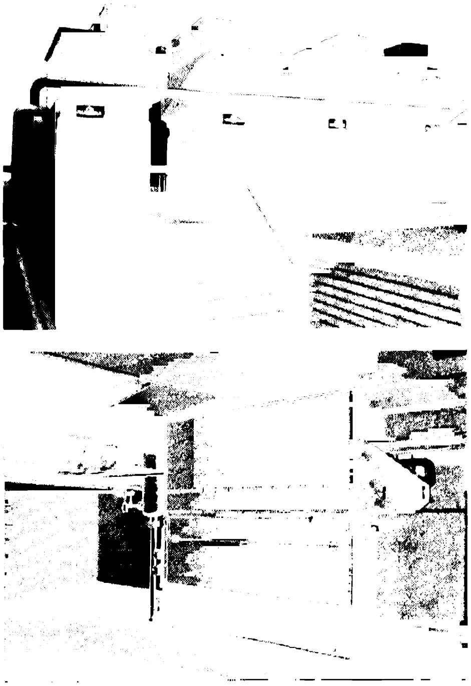
NIAGARA FAN COOLERS— Bulletin #72, Part 2

NIAGARA AEROPASS CONDENSERS— Bulletin #111

NIAGARA AERO HEAT EXCHANGER— Bulletin #96

NIAGARA STANDARD COIL SECTIONS—Bulletin #92

NIAGARA MOTOR BLOWERS— Bulletin #89, Part 1



NIAGARA AEROPASS CONDENSERS

Refrigeration plant operators, using the Niagara Aeropass Condenser, gain as much as 35% of power by reducing compressor head pressures. This gain is assured permanently because the patent Duo-Pass prevents scale depositing on Condenser tubes. Equipped with the Niagara "Oil-out," a system free from oil is obtained.

NIAGARA "No-Frost" METHOD

For refrigerated rooms below freezing, the Niagara "No-Frost" Method gives always full capacity at lower cost because no ice ever forms on coils. It provides rapid, uniform cooling for products, trouble-free operation and lower operating expenses.

EXPANSION VALVES, HAND OR NEEDLE

(A—Ammonia; B—Other refrigerants)

Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
(A) **Baker Refrigeration Corp., S. Windham, Me.** *(p. 63)*
Crane Co., 836 S. Michigan Ave., Chicago 5, Ill. *(p. 109)*
(A) **Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill.** *(p. 49)*
Darling Valve & Mfg. Co., Foot of Walnut St., Williamsport, Pa.
(A,B) **Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill.** *(p. 46)*
(A,B) **Frick Co., Waynesboro, Pa.** *(p. 51)*
(A,B) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
(A,B) **Henry Valve Co., Melrose Park, Ill.** *(p. 203)*
(B) **Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa.** *(p. 205)*
(A) **Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal.** *(p. 39)*
(B) Mueller Brass Co., Port Huron, Mich.
Heco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Philadelphia 46, Pa.
(B) **Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa.** *(p. 204)*
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y
(A,B) **Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis.** *(p. 33)*
(A) Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
(A,B) **Watson-Stillman Co., Roselle, N.J.** *(p. 111)*
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A,B) **Worthington Pump & Machinery Corp., Harrison, N.J.** *(p. 144)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) **York Corp., York, Pa.** *(p. 52)*

EXPANSION VALVES, SPECIAL LOW TEMPERATURE

(A—Ammonia; B—Other refrigerants)

(A,B) **Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo.** *(p. 99)*
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
(B) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
(A,B) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
(A,B) Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(B) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17 Mo.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

EXPANSION VALVES, THERMOSTATIC

(A—Ammonia; B—Methyl Chloride; C—Sulfur; D—Freon)

(A,B,C,D) **Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo.** *(p. 99)*
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
(B,C,D) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
(B,C,D) **Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O.** *(p. 17)*
(B,C,D) **General Controls Co., 801 Allen Ave., Glendale 1, Cal.** *(p. 64)*
(B,C,D) Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.
(A,B,C,D) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
(B,D) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B,D) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

FANS, FAN BLADES, etc., (See also BLOWERS)

Acme Equip. Co., 205 E. Broadway, Muskogee, Okla.
Advance Fan & Blower Co., 3428 Bagley, Detroit, Mich.
Aerovent Fan Co., 710 E. Ash St., Piqua, O.
Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aire-Foile Fan & Blower Co., 2909 Central, Detroit 9, Mich.
Airmaster Corp., 4317 Ravenswood Ave., Chicago 13, Ill.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Allen Ventilator Div., Production Planning Co., 704 Woodward, Rochester, Mich.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Coolair Corp., 3606 Mayflower St., Jacksonville, Fla.
American Machine Products Co., Marshalltown, Ia.
Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Bauer Bros Co., 1774 Sheridan Ave., Springfield, O.
Bayley Blower Co., 1817 S. 66th St., Milwaukee 14, Wis.
Thomas Beckett & Co., Inc., P.O. Box 7354, Dallas 2, Tex.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
Burt Mfg. Co., 932 S. High St., Akron 11, O.
C & H Air Conditioning Fan Co., Inc., 1603 DeKalb Ave., N.E., Atlanta, Ga.
Champion Blower & Forge Co., Harrisburg & Charlotte Sts., Lancaster, Pa.
Chelsea Fan & Blower Co., Inc., 639 South Ave., Plainfield, N.J.
Chicago Blower Corp., 4588 W. Congress St., Chicago 24, Ill.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
De Bothezat Fans, Div. of American Machine & Metals, Inc., E. Moline, Ill.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Dual-Air Fan Co., P.O. Box 169, S. Elgin, Ill.
Duriron Co., Inc., Dayton 1, O.
Eagle-Picher Sales Co., American Bldg., Cin'ti 1, O.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
Fasco Industries, Inc., Union & Augusta Sts., Rochester 2, N.Y.
Garden City Fan Co , 332 S. Michigan Ave., Chicago 4, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y
General Industries Co., Olive & Taylor St., Elyria, O.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Joy Mfg. Co., Oliver Bldg., Pittsburgh 22, Pa.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
King Co., 902 N. Cedar St., Owatonna, Minn.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks. Inc., Allentown, Pa.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Maurey Mfg. Corp., 2907 S. Wabash Ave., Chicago 16, Ill.
Meier Elec. & Machine Co., 3525 E. Washington, Indpls. 7, Ind.
Moore Co., Marceline, Mo.
National Engrg. & Mfg. Co., 519 Wyandotte St., Kansas City 6, Mo.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Newcomb-Detroit Co., 5741 Russell St., Detroit 11, Mich.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
Northern Blower Co., W. 65th St., South of Denison, Cleveland 2, O.
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.

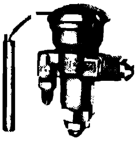
(Continued)

Alco Valve Company

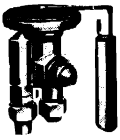
ENGINEERED REFRIGERANT CONTROLS
851 Kingsland Avenue, St. Louis 5, Mo.
NEW YORK OFFICE: 55 WEST 42ND STREET
Chicago Office: 4534 North Broadway

THE COMPLETE LINE OF REFRIGERANT CONTROLS

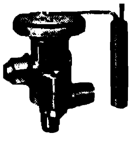
ALCO THERMO EXPANSION VALVES: for automatic control of liquid refrigerant on all types of refrigeration and air conditioning systems. Capacities—from fractional tonnage to 50 tons "Freon-12," 100 tons Methyl Chloride. Low temperature valves for —40° F. to —100° F.



Type 402 with pressure limiting feature



Type TK—"3 valves in 1"



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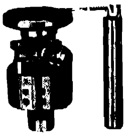
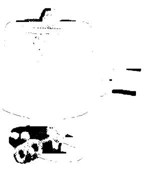


Type TR—Multi-Outlet

ALCO SOLENOID VALVES: for all types of service. For Liquid: "Freon"—up to 75 tons. Methyl Chloride—up to 150 tons. For Suction: "Freon"—up to 8.8 tons. Methyl Chloride—up to 17 tons. For brine, water, gas, air and steam.



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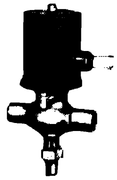
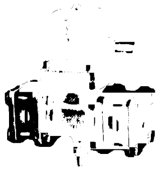
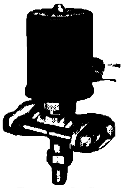


Type M3

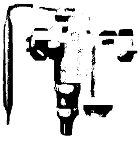


Type R2

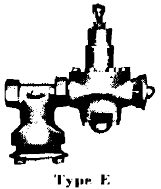
ALCO AMMONIA CONTROLS: Solenoid Liquid Valves—up to 172 tons. Solenoid Suction Valves—up to 28 tons. Thermo Expansion Valves—from fractional tonnage to 125 tons. Automatic Expansion Valves—from fractional tonnage to 60 tons.



Type M9F



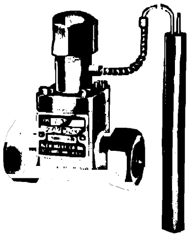
Type UC



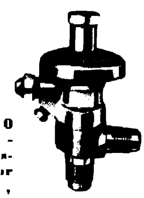
Type E with Strainer

ALCO SUCTION LINE CONTROLS:

Type EPR—for all refrigerants, with connection sizes up to 6 in.



Type 732 SNAP-ACTION SUCTION VALVE—Temperature operated—½ ton, "Freon-12"—1 ton, Methyl Chloride.



Type 760 "EVAPO-TROL"—Pressure Regulator—½ ton, "Freon-12"—¾ ton, Methyl Chloride.

ALCO also makes: Constant Pressure Expansion Valves—Liquid and Suction Line Strainers.

FANS (Continued)

Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Piqua Machine & Mfg. Co., Young & College Sts., Piqua, O.
J. F. Pritchard & Co., 908 Grand Ave., Kansas City 6, Mo. *(p. 72)*
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Robbins & Myers, Inc., 1345 Lagonda Ave., Springfield, O.
Robinson Ventilating Co., Zelienople, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
V. E. Sprouse Co., Inc., Columbus, Ind.
Standard Elec. Mfg. Co., W. Berlin, N.J.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 28)*
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Swift Mfg. Co., Inc., 1455 E. Nine Mile Rd., Hazel Park, Mich.
Torrington Mfg Co., 70 Franklin St., Torrington, Ct. *(p. 27)*
Trane Co., La Crosse, Wis. *(p. 118)*
Transaire Mfg. Co., Div. of Penn Elec. Motor Co., 1825 Wylie Ave., Phila. 30, Pa. *(p. 100)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
U. S Thermo Control Co., 44 S. 12th St., Minneapolis 4, Minn.
Utility Appliance Corp., 4851 S. Alameda St., Los Angeles 11, Cal.
Victor Elec. Products, Inc., 2950 Robertson Rd., Cincinnati 9, O.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

FANS, CONDENSER

Meier Elec. & Machine Co., Inc., 3525 E. Washington St., Indianapolis 7, Ind.
Transaire Mfg. Co., Div. of Penn Elec. Motor Co., 1825 Wylie Ave., Phila. 30, Pa. *(p. 100)*
Swift Mfg. Co., Inc., 1455 E. 9 Mile Rd., Hazel Park, Mich.

FAN COIL UNITS (See AIR CONDITIONING FAN COIL UNITS)

FARM FREEZERS (See HOME & FARM FREEZERS)

FASTENERS & FASTENING DEVICES (See also BOLTS, NUTS, SCREWS, etc.)

Autoscrew Co., 216 W. 18th St., N.Y.C 11
Bristol Co., Waterbury 20, Ct.
Buffalo Bolt Co., 101 East Ave., N. Tonawanda, N.Y.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Cherry Rivet Co., 231 Winston St., Los Angeles 13, Cal.
Continental Screw Co., 459 Mt. Pleasant St., New Bedford, Mass.
Crescent Belt Fastener Co., 480 Lexington Ave., N.Y.C. 17
Dzus Fastener Co., Inc., Babylon, N.Y.
B. F Goodrich Co., 500 S. Main St., Akron, O.
H. M. Harper Co., 8200 Lehigh Ave., Morton Grove, Ill.
Lexington Supply Co., 4815 Lexington Ave., Cleveland 3, O.
Miracle Adhesives Corp., 214 E. 53rd St., N.Y.C. 22
Goodloe E. Moore Co., 2811 N. Vermilion, Danville, Ill.
National Lock Co., 7th St., & 18th Ave., Rockford, Ill.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Prestole Corp., 1345 Miami St., Toledo, O.
Rockford Screw Products Co., 2501-9th St., Rockford, Ill
St. Louis Screw & Bolt Co., 6900 N. B'way, St. Louis 15, Mo.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Shakeproof, Inc., 2501 N. Keller Ave., Chicago 39, Ill.
Simmons Fastener Corp., N. Broadway, Albany 1, N.Y
South Chester Corp., 1411 Finance Bldg., Philadelphia 2, Pa. *(p. 101)*

A FAST, ECONOMICAL REPLACEMENT FOR HERMETIC UNIT CONDENSER FAN ASSEMBLIES

TRANSAIRE
MOTOR FAN BRACKET ASSEMBLY

The TRANSAIRE MFB is designed to replace condenser fan assemblies found on most hermetically sealed refrigeration units . . . regardless of make or manufacturer. It eliminates relying upon hermetic unit manufacturers when a condenser fan needs replacing.

★ PERMITS USE OF ORIGINAL MOUNTING HOLES WITHOUT REFERENCE TO MODEL OR CATALOG NO.
★ ADJUSTABLE LEGS FOR EASY CENTERING
★ IMPELLER FAN AVAILABLE IN 4 SIZES
★ DUST-PROOF MOTOR, RUBBER MOUNTED
★ INSTALLED BY ANY QUALIFIED SERVICEMAN

GUARANTEED
FOR
ONE YEAR

Ask your Refrigeration Supply House or write us for details

PENN ELECTRIC MOTOR COMPANY
1825 WYLIE STREET—PHILADELPHIA 30, PENNSYLVANIA

Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 13, O.
Townsend Co., 205 River Rd., New Brighton, Pa.
Triplex Screw Co., 5317 Grant Ave., Cleveland 4, O.
United-Carr Fastener Corp., 31 Ames St., Cambridge 42, Mass.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20,

FAUCETS, BEER, CARBONATED WATER, etc.

Hudson Products Co., Inc., 440 St. Aubin, Detroit 7, Mich.
Multiplex Faucet Co., 4325 Duncan Ave., St. Louis 10, Mo.
Penn Engrg. & Mfg. Corp., 30 S. Main St., Doylestown, Pa.
R- Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p. 89)*

FAUCETS, DRINKING WATER (See DRINKING WATER COOLER FITTINGS)

FELT (See also INSULATION)

American Felt Co., Glenville, Ct.
American Hair & Felt Co., 222 N. Bank Dr., Chicago 54, Ill. *(p. 134)*
Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Grant Wilson, Inc., 315 S. Sherman, Chicago 4, Ill.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 139 E. 19th St., N.Y.C. 11

FERMENTATION ROOMS, BAKERY

Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O.
Tenney Engrg., Inc., 26 Ave. B, Newark, N.J.
Union Steel Products Co., 448 Pine St., Albion, Mich. *(p. 175)*

FIBRE, VULCANIZED, SHEET; ROD; TUBE, etc.

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Continental Diamond Fibre Co., 3 Chapel St., Newark, Del.
Taylor Fibre Co., Norristown, Pa.
Technical Ply-Woods, 228 N. La Salle St., Chicago 1, Ill.
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.
Grant Wilson, Inc., 316 S. La Salle St., Chicago 4, Ill.

FILTERS (See particular type; also STRAINERS; also SCALE TRAPS)

FILTERS, AIR (See AIR FILTERS)

FILTERS, AIR LINE, OIL, WATER, etc. (See also STRAINERS)

Air Maze Corp., 5200 Harvard Ave., Cleveland 5, O.
American Water Softener Co., S.E. Corner 4th & Lehigh Ave., Phila. 33, Pa.
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Bowser, Inc., 1302 E. Creighton Ave., Ft. Wayne 2, Ind.
Buffalo Wire Works, 3200 Terrace, Buffalo 3, N.Y.
Buhring Water Purifying Co., 4 W. Garfield Ave. Atlantic Highlands, N.J.
Burt Mfg., Co., 932 S. High St., Akron 11, O.
Calgon, Inc., Hagan Bldg., Pittsburgh 30, Pa.
Carborundum Co., P.O. Box 337, Niagara Falls, N.Y.
Chicago Filter Co., Joliet, Ill.
Cochrane Corp., 17th St below Allegheny Ave., Phila. 32, Pa.
Conoflow Corp., 2100 Arch St., Phila. 3, Pa.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
De Vilbiss Co., 300 Phillips Ave., Toledo 1, O.
Dollinger Corp., 1 Centre Park, Rochester 3, N Y.
Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.
Elgin Softener Corp., Elgin, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 90)*
Fischer & Porter Co., Hatboro, Pa.
Fram Corp., 55 Pawtucket Ave., Providence 16, R.I.
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11, N.J.
Graver Water Conditioning Co., 216 W. 14th St., N.Y.C. 11
Greer Hydraulics, Inc., 454-18th St., Brooklyn 15, N.Y.
Henry Valve Co., Melrose Park, Ill. *(p. 102)*
Honan-Crane Corp., Div. of Houdaille-Hershey Corp., 64 Indiana Ave., Lebanon, Ind.
Hugerford & Terry, Inc., 226 Atlantic Ave., Clayton, N.J.
Hugo Mfg. Co., 49 Ave., W. & Superior St., Duluth 7, Minn.
Infilco, Inc., 2750 S. 12th St., Tucson, Ariz.
Logan Engrg. Co., 4801 W. Lawrence, Chicago 30, Ill.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
C. A. Norgren, 222 Santa Fe Drive, Denver, Col.
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Oshkosh Filter & Softener Co., 51 Ceape St., Oshkosh, Wis.
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.
Pacific Lumber Co., By Products, 100 Bush St., San Francisco 4. Cal. *(p. 137)*

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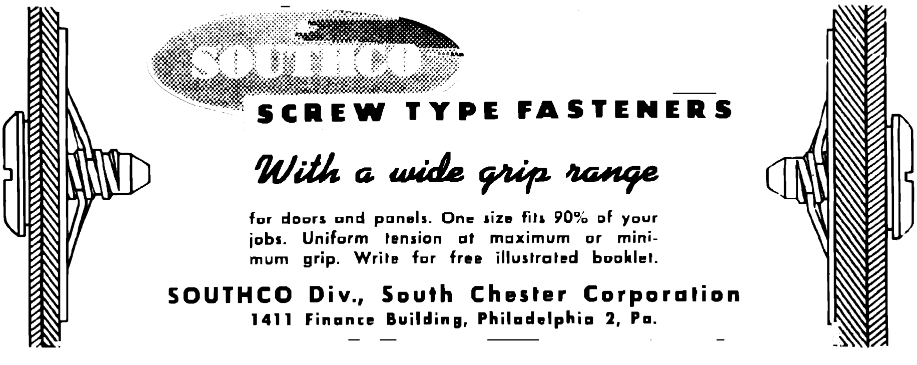
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SCREW TYPE FASTENERS
With a wide grip range
for doors and panels. One size fits 90% of your jobs. Uniform tension at maximum or minimum grip. Write for free illustrated booklet.
SOUTHCO Div., South Chester Corporation
1411 Finance Building, Philadelphia 2, Pa.

FILTERS, AIRLINE (Continued)

Puro Filter Corp. of America, 440 Lafayette St., N.Y.C. 3
Reso Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Schwartz Mfg. Co., Two Rivers, Wis.
Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
Sunroc Co., Glen Riddle, Pa. *(p. 89)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N. Y.
W.G.B. Oil Clarifier, Inc., Kingston, N.Y.
Wayne Brass Co., Inc., 230 W. Main St., Fort Wayne, Ind.
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.

FILTERS, MAGNETIC

Fischer & Porter Co., Hatboro, Pa.
Stearns Magnetic Mfg. Co., 667 S. 28 St., Milwaukee 4, Wis.

FILTERS, REFRIGERANT (See also STRAINERS)

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 149)*
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Baker Refrigerator Corp., S. Windham, Me. *(p. 63)*
Cee-Kleer Products, Inc., 947 W. 6th St., Cincinnati 3, O.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Henry Valve Co., Melrose Park, Ill. *(p. 102)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Remco, Inc., Zelienople, Pa. *(p. 87)*
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 88)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.

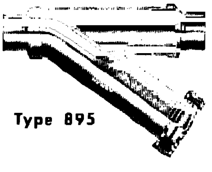
FINISHES

Acme White Lead & Color Wks., 8250 St. Aubin Ave., Detroit, Mich.
American Chemical Paint Co., Ambler, Pa.
American Marietta Co., 43 E. Ohio St., Chicago, Ill.
Arco Co., 7301 Bessemer Ave., Cleveland 27, O.
Atlas Powder Co., N. Chicago, Ill.
Baer Bros., 438 W. 37th St., N.Y.C. 18
Chicago Vitreous Enamel Products Co., 1401-47 S. 55th Court, Cicero 50, Ill.
Dennis Chemical Co., 2701 Papin St., St. Louis 3, Mo.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y
Glidden Co., 11001 Madison Ave., Cleveland 2, O.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Hercules Powder Co., Delaware Trust Bldg., Wilmington, Del.
A. C. Horn Co., Inc., 10th St. & 44th Ave., Long Island City 1, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Interchemical Corp., 57 State St., Newark, N.J.
Jones-Dabney Co., Div. of Devoe & Raynolds, 1481 S. 11th St., Louisville 8, Ky.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Master Mechanics Co., Freeman Rd., Cleveland 13, O.
Monsanto Chemical Co., 600 Monsanto Ave., Springfield 2, Mass.
National Lead Co., 111 Broadway, N Y.C. 6
O'Neil Duro Co., P.O. Box 1166, Milwaukee 1, Wis.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Randolph Products Co., Carlstadt, N.J. *(p. 103)*
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y.C. 16
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
United Chromium, Inc., 100 E. 42nd St., N.Y.C. 17
U S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J
Zapon Div., Atlas Powder Co., Stamford, Ct.

FINNED COILS

(A—Ammonia; B—Other refrigerants)

(A,B) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
(A,B) Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
(B) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B) Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*



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(B) Blissfield Mfg. Co., 626 Depot St., Blissfield, Mich.
Brown Fintube Co., Elyria, O.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct *(p. 40)*
(A,B) Conditionaire Unit Co., Div. of Howe Ice Machine Co., 2815 Montrose Ave., Chicago 18, Ill. *(p. 198)*
(A) Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.
(B) Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
(B) Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y. *(p. 59)*
(A,B) Frick Co., Waynesboro, Pa. *(p. 51)*
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(B) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
(A,B) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(B) Hastings Air Conditioning Co., Inc., Hastings, Neb.
(A,B) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
(B) Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 88)*
(A,B) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
(A,B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. *(p. 6)*
(A,B) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 59)*
(A,B) Kramer-Trenton Co., Olden & Bruening Aves., Trenton 5, N.J. *(p. 195)*
Larkin Coils 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(B) McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. *(p. 55)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
(B) Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N. J.
(B) Reese & Long Refrigeration Products, Inc., 1891 Park Ave., N.Y.C. 35
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(B) Rome-Turney Radiator Co., Rome, N.Y. *(p. 105)*
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona Cal.
(A,B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B. Newark 5, N.J.
(B) Trane Co., La Crosse, Wis. *(p. 118)*
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. *(p. 37)*
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 52)*
Young Radiator Co., Racine, Wis. *(p. 77)*

FINNED TUBING

Aerofin Corp., 410 Geddes St., Syracuse N.Y. *(p. 11)*
American Coils Co., 25 Lexington St., Newark 5, N.J.
Brown Fintube Co. Elyria, O.
Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
Detroit Ice Machine Co., 2615-12th St., Detroit, Mich.

(Continued)

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Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Heintz Mfg. Co., Front St & Olney Ave., Phila. 20, Pa.
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Kramer-Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. (*p. 55*)
Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N.J.
Refrigeration Appliances, Inc., 923 Lake St., Chicago, Ill. (*p. 201*)
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
Rome-Turney Radiator Co., Rome, N.Y. (*p. 105*)
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Tilco Fin Tubing Div, Kentile, Inc., 58 2nd Ave, Brooklyn 15, N Y
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co, 1411 Central Ave., Detroit 9, Mich
Young Radiator Co., Racine, Wis. (*p. 77*)

FISH REFRIGERATORS (See DISPLAY CASES)

FITTINGS, AMMONIA

Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Capitol Mfg & Supply Co, 153 W. Fulton St, Columbus O.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Henry Valve Co., Melrose Park, Ill. (*p. 104*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. (*p. 155*)
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky
Henry Vogt Machine Co, 10th & Ormsby St., Louisville 10, Ky.
Watson-Stillman Co., Roselle, N.J. (*p. 111*)
Worthington Pump & Machinery Corp., Harrison, N.J (*p. 144*)
XL Refrigerating Co, 1834 W. 59th St., Chicago 36, Ill
York Corp., York, Pa. (*p. 54*)

FITTINGS, BRAZING & SOLDERING

Capitol Mfg & Supply Co., 153 W. Fulton St, Columbus, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 203*)
Mueller Brass Co 1925 Lapeer Ave, Port Huron, Mich
National Lead Co., 111 Broadway, N.Y.C. 6
Northern Indiana Brass Co, 935 Plum St., Elkhart, Ind
United Wire & Supply Corp 1197 Elmwood Ave, Providence 7, R I
Walworth Co, 60 E. 42nd St., N Y.C. 17
Watson-Stillman Co., Roselle, N.J. (*p. 111*)
Wayne Brass Co, Inc, 230 W Main St., Fort Wayne, Ind.

FITTINGS, CAPILLARY TUBE

Kenmore Machine Products Inc, 15 Depew Ave Lyons, N Y.
Mueller Brass Co, 1925 Lapeer Ave Port Huron Mich
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill

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Frick Co., Waynesboro, Pa. (*p. 51*)
Kuhns Bros. Co., 1822 McCall St., Dayton 1, O.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Vibraseal Corp., 2832 E. Grand Blvd., Detroit 11, Mich.
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis.
Watson-Stillman Co., Roselle, N.J. (*p. 111*)
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

FITTINGS, CAST IRON

American Flexible Coupling Co., 1808 Pittsburgh Ave., Erie, Pa.
James B. Clow & Sons, 201 N Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. (*p. 51*)
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Grinnell Co., 260 W. Exchange St., Providence 1, R.I.
Henry Valve Co., Melrose Park, Ill. (*p. 104*)
Kennedy Valve Mfg. Co., Elmira, N.Y.
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. (*p. 83*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
J. P. Ward Foundries, Inc., Blossburg, Pa.
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
XL Refrigerating Co., 1834 W 59th St., Chicago 36, Ill.

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Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
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FITTINGS, ELECTRIC CONDUIT (See also CLAMPS, CONDUIT & CABLE)

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Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
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General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
Mineralloc Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
National Elec. Products Corp., Chamber of Commerce Building, Pittsburgh 19, Pa.
Pyle Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill.
Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12, Wis.

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Armco Drainage & Metal Products, Inc., Middletown, O.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. (*p. 51*)
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Henry Valve Co., Melrose Park, Ill. (*p. 104*)
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Kuhns Bros Co., 1822 McCall St., Dayton 1, O.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Phoenix Mfg. Co., Joliet, Ill.
Standard-Keil Hardware Mfg. Co., Inc., 639 Broadway, N.Y.C. 12
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. (*p. 83*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
Watson-Stillman Co., Roselle, N.J. (*p. 111*)
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa (*p. 52*)

FITTINGS, FLARE, REFRIGERANT

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
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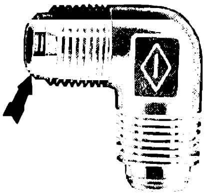

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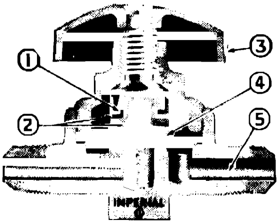

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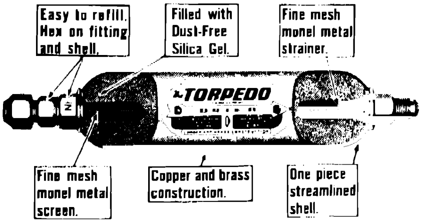
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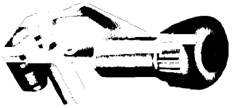

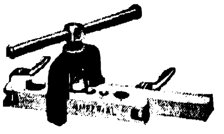

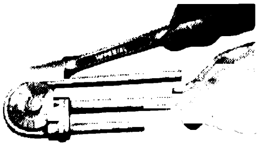

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FITTINGS, FLARE, REFRIGERANT (Continued)

Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Imperial Brass Mfg. Co., 437 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Madden Brass Products Co., 111 N. Franklin St., Chicago 10, Ill.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Remco, Inc., Zellenople, Pa. *(p. 106)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburg 26, Pa. *(p. 104)*
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.
Watson-Stillman Co., Roselle, N.J. *(p. 111)*
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.

FITTINGS, FORGED

Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus O.
Commonwealth Brass Corp., 5835 Commonwealth Ave., Detroit 8, Mich.
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 9, Ill. *(p. 46)*
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Graver Tank & Mfg. Co., Inc , 4809 Tod Ave., E. Chicago 1, Ind.
Henry Valve Co., Melrose Park, Ill. *(p. 104)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Phoenix Mfg. Co., Joliet, Ill.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 104)*
Taylor Forge & Pipe Wks., P.O. Box 485, Chicago 90, Ill.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 111)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

FITTINGS, HARD RUBBER

American Hard Rubber Co., 11 Mercer St , N.Y.C. 13
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5 N.Y.
Luzerne Rubber Co., Trenton 9, N.J.
NAtional Motor Bearing Co., Inc., Redwood City, Cal.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
Thermoid Co., Trenton, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N Y C. 20

FITTINGS, HOSE (See also HOSE CLAMPS)

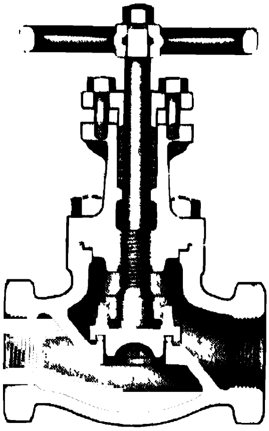
Aeroquip Corp., 300 S. East Ave., Jackson, Mich.
Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Boston Woven Hose & Rubber, P.O. Box 1071, Boston 3, Mass.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Flexible Tubing Corp., New Whitfield St., Guilford Ct.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
C. A. Norgren, 222 Santa Fe Drive, Denver, Col.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Thermoid Co., Trenton, N.J.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 31, O.
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.

FITTINGS, PIPE

Allmetal Screw Products Co., Inc., 33 Greene St., N Y.C 13
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Armco Drainage & Metal Products, Inc., Middletown, O.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Catawissa Valve & Fittings Co., Catawissa, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill. (Sanitary)
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Commonwealth Brass Corp., 5835 Commonwealth Ave , Detroit 8, Mich. (Brass only)
Cooper Alloy Foundry Co., Bloy St. & Ramsey Ave., Hillside 5, N.J.
Crane Co., 836 Michigan Ave., Chicago 5, Ill *(p. 109)*
Dart Union Co , 134 Thurbers Ave., Providence 5, R.I.
Defiance Automatic Screw Co., Ft. Wayne Rd., Defiance, O.
Dersch, Gesswein & Neuert, Inc., 4845 W Grand Ave., Chicago 39, Ill. *(p. 46)*
Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Duriron Co., Inc., Dayton 1, O.
Ellsworth Pipe & Supply Co., 513 W. Traser St., Milwaukee 3, Wis.
Everhot Products Co., 2001 Carroll Ave , Chicago 12, Ill.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 51)*
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 104)*
Holo-Krome Screw Corp., 25 Brook St., Hartford 10, Ct.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Jefferson Union Co., 601 W. 26th St., N.Y.C. 1.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Kuhns Bros Co., 1822 McCall St., Dayton 1, O.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti 14, O.
Madden Brass Products Co., 1111 N. Franklin St., Chicago 10, Ill.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Minerallac Elec Co., 25 N. Peoria St., Chicago 7, Ill
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
A. B. Murray Co., Inc , 630 Green Lane, Elizabeth, N.J.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Reeves Steel & Mfg. Co , 137 E. Iron Ave , Dover, O.
Reynolds Metals Co., 2000 S. 9th St., Louisville, Ky.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
M. B. Skinner & Co., South Bend 23, Ind.
Set Screw & Mfg. Co., 75 Main St., Bartlett, Ill.
Taylor Forge & Pipe Wks., P.O. Box 485, Chicago 90, Ill.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Victaulic Co. of America, P.O. Box 509, Elizabeth, N.J.
Henry Vogt Machine Co., 10th & Ormsby Sts., Louisville 10, Ky.
Walworth Co., 60 E 42nd St., N.Y.C. 17
J. P. Ward Foundries, Inc., Blossburg, Pa.

(Continued)

FOR ALL REFRIGERATION PIPING NEEDS ... CHECK THE COMPLETE CRANE LINE



No. 1504, Globe. Working Pressure: 300 pounds ammonia. Sizes: ¼ to 3 inch.

SAFER CONTROL OF REFRIGERANTS Crane Ferrosteel Globe and Angle Valves are especially designed for high pressure ammonia refrigeration service. Special Crane seating design, rugged all-iron construction, and precise manufacture make these valves safer, more durable and more efficient when handling deadly or hazardous gases.

To assure tightness in service, Crane Special Lead face disc seats securely upon a crowned face in the body. A machined "back seating" shoulder on the stem relieves internal pressure on packing, insures pressure-tight joint, when valve is wide open. Flanged valves have tongue and groove faces as an added protection against gasket blowout; screwed valves have long threads and are recessed for soldering. See your Crane Catalog.

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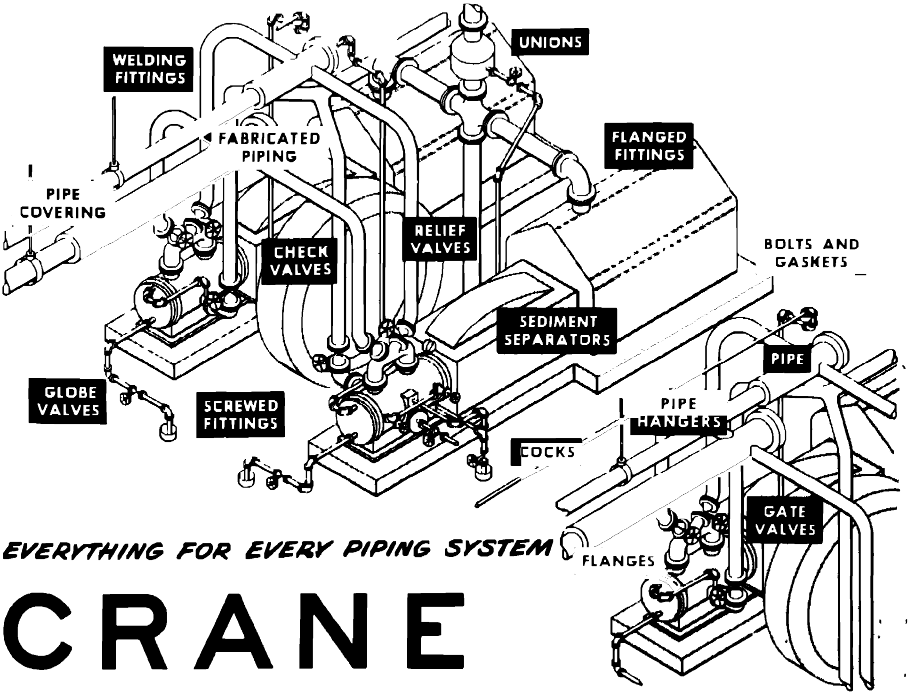

EVERYTHING FOR EVERY PIPING SYSTEM

CRANE

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FITTINGS, PIPE (Continued)

Watson-Stillman Co., Roselle, N.J. *(p. 111)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*
J. A. Zurn Mfg. Co., Erie, Pa.

FITTINGS, SHEET METAL (See DUCT ACCESSORIES; also DUCTWORK, PREFABRICATED)

FITTINGS, WELDING

Bonney Forge & Tool Wks., Allentown, Pa.
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Cooper Alloy Foundry Co., Bloy St. & Ramsey Ave., Hillside 5, N.J.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Dresser Mfg. Div., 490 Fisher Ave., Bradford, Pa.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 51)*
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Taylor Forge & Pipe Wks, P.O. Box 485, Chicago 90, Ill.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 111)*

FLAKE ICE MACHINERY (See ICE MAKING MACHINES)

FLANGE ADAPTERS (See ADAPTER FLANGES)

FLANGES, PIPE (See FITTINGS)

FLEXIBLE CONNECTIONS (See also CHARGING LINES; also TUBING, FLEXIBLE; also SWING JOINTS; also VIBRATION ABSORBERS, LINE)

American Brass Co., Waterbury 88, Ct. *(p. 189)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Atlantic Metal Hose Co., Inc., 123 W. 64th St., N.Y.C. 23
Barco Mfg. Co., 1801 Winnemac Ave., Chicago 40, Ill.
Chicago Metal Hose Corp., Maywood, Ill.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Flexible Tubing Corp., New Whitfield St., Guilford, Ct.
Henry Valve Co., Melrose Park, Ill. *(p. 104)*
Johnson Metal Hose Co., 226 Mill St., Waterbury 88, Ct.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Mayson Mfg. Co., 4332 Horatio St., Detroit 10, Mich.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y.
Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
Uniflex Metal Hose Co., 52 Rubber Ave., Naugatuck, Ct.
York Corp., York, Pa. *(p. 52)*

FLEXIBLE COUPLINGS (See also SHAFT COUPLINGS)

Ajax Flexible Coupling Co., Inc., Westfield, N.Y.
American Flexible Coupling Co., 1808 Pittsburgh Ave., Erie, Pa.
Baldwin Duckworth Div., Chain Belt. Co., 369 Plainfield St., Springfield, Mass.
Boston Gear Works, Quincy, Mass.
Bushings, Inc., 4358 Collidge Highway, Royal Oak, Mich.
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Centric Clutch Co., Woodbridge, N.J.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Congress Drive Div., Tann Corp., 3750 E. Outer Drive, Detroit 34, Mich.
Delavan Mfg. Co., 3009 6th Ave., Des Moines 13, Ia.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago 24, Ill.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind. (Shaft)
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa. (For power Transmission)
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Morse Chain Co., a Borg-Warner Industry, Ithaca, N.Y.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
W. H. Nicholson & Co., 209 Oregon St., Wilkes-Barre, Pa.
Palmer-Bee Co., 1761 Poland Ave., Detroit 12, Mich.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 43, Pa.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.
Spicer Mfg. Co., 4100 Bennett Rd., Toledo 1, O.
Uniflex Metal Hose Co., 52 Rubber Ave., Naugatuck, Ct.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7 (Pipe)
Westinghouse Elec. Corp., 200 McCandless Ave., Pittsburgh 1, Pa.

FLOAT BALLS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 178)*
Chicago Float Wks., Inc., 2330 S. Western Ave., Chicago 8, Ill.
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Industrial Wire Cloth Products Corp., Wayne, Mich. *(p. 88)*
W. H. Nicholson & Co., 209 Oregon St., Wilkes-Barre, Pa.
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Amos Thompson Corp., Edinburg, Ind.

FLOATS, LIQUID LEVEL CONTROL (See FLOAT SWITCHES; also HIGHSIDE FLOATS; also LOWSIDE FLOATS; also CONTROLS, LIQUID LEVEL)

FLOAT SWITCHES (See also HIGHSIDE FLOATS; also LOWSIDE FLOATS)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
Allen-Bradley Co., Milwaukee 4, Wis.
Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Fischer & Porter Co., Hatboro, Pa. (Flow Alarm)
Frick Co., Waynesboro, Pa. *(p. 51)*
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Jo-Bell Products, Inc., 4840 South St. Louis, Chicago, Ill.
McDonnell & Miller, Inc., 3500 N. Spaulding Ave., Chicago 18, Ill.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Mojonnier Bros. Co., 4601 W. Ohio St., Chicago, Ill.
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Philadelphia 46, Pa.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.

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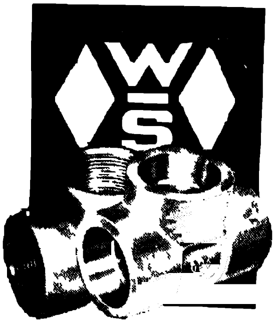
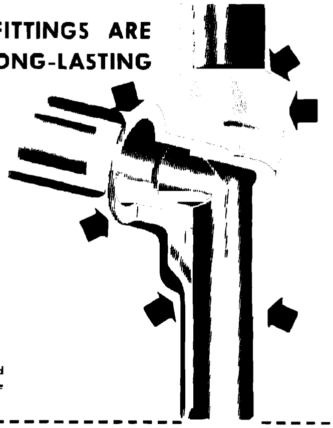


The Ideal Light-weight Screw End and Socket Weld Fittings for . . . GENERAL REFRIGERATION SERVICE

Precision machined from solid—carbon, stainless and alloy steel—drop forgings They're built right to fit tight . permanently They cut maintenance costs

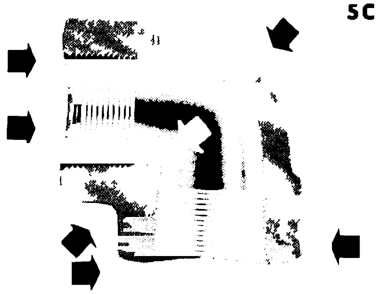
SOCKET WELDING FITTINGS ARE EASY TO INSTALL, LONG-LASTING



- Just slip fitting over end of pipe and weld
- Deep sockets support and align pipes eliminate tack welding the use of special fixtures accurate measuring and cutting of pipe Ample come and go
- Position of weld prevents notches Wall sections 1½ times nominal pipe thickness permit proper heat penetration correctly proportioned fillet weld
- Inside diameter of fitting matches inside of pipe for smooth flow
- No weak sections—long bands extend well beyond bottom of socket provide reinforcement at points of strain
- Joints are extra strong and leak proof unaffected by vibration distortion or shock You can put these fittings in and forget them

All are available for use with Schedule 40 Standard Weight Pipe Schedule 80 Extra Strong Pipe Schedule 160 and Double Extra Strong Pipe Sizes ⅛ to 4 I P S

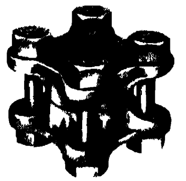
SCREW-END TYPES FOR ALL PRESSURE AND TEMPERATURE SERVICES



- Threads are long accurately cut
- Designed for perfect alignment
- Uniform wall thickness
- Unrestricted flow
- Controlled chemical and physical properties
- Long low bands afford better wrench grip
- Rugged bored from solid drop forgings

9-K-18

• FORGED STEEL FLANGE UNIONS •



Two and Four Bolt Type especially designed for Refrigeration Service

Screw and Socket Weld Types

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DISTRIBUTOR PRODUCTS DIV **WATSON-STILLMAN** ROSELLE, NEW JERSEY

Designers and Manufacturers of Forged Steel Fittings Valves Wire Rope Shears Hand Pumps Jacks Pipe Benders and Hydraulic Equipment

FLOAT SWITCHES (Continued)

Soreng Mfg. Corp., 9555 Eden Ave., Schiller Park, Ill.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. (*p. 89*)
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

FLOAT VALVES, REFRIGERANT (See HIGHSIDE FLOATS; also LOWSIDE FLOATS)

FLOAT VALVES, WATER

Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Climax Controls Div. of Black, Sivalls & Bryson, Inc., 15 N. Cincinnati, Tulsa, Okla.
Davis Regulator Co., 2511 S Washtenaw Ave., Chicago 8, Ill.
Farm Appliances, Inc., Sturgeon Bay, Wis.
Flint & Walling Mfg. Co., Inc., 95 Oak St., Kendallville, Ind.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Klipfel Valves, Inc., 1000 Weller Ave., Hamilton, O.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
McDonnell & Miller, Inc., 3500 N. Spaulding Ave., Chicago 18, Ill.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Robert Mfg. Co., 9035 Venice Blvd., Los Angeles 34, Cal.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Trane Co., La Crosse, Wis. (*p. 118*)
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

FLORISTS REFRIGERATORS (See REFRIGERATORS, FLORISTS)

FLOW CONTROLS (See REGULATORS, FLOW)

FLOW RATE INDICATORS

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., P.O. Box 1342, Providence 1, R.I.
Fischer & Porter Co., Hatboro, Pa.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I
Hastings Instrument Co., Inc., Hampton, Va.
Hays Corp., Michigan City, Ind.
Henszey Co., 202½ N. Water St., Watertown, Wis.
Infilco, Inc., 2750 S. 12th Ave., Tucson, Ariz.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

FLYWHEELS

Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago 24, Ill.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 89*)
Linderme Machine & Tool Co., 12241 Coyle Ave., Detroit 27, Mich.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.

FOAM RUBBER (See RUBBER PRODUCTS, CELLULAR)

FOIL, METALLIC

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Aluminum Co. of America, Pittsburgh 19, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.

FOOT VALVES (See VALVES, FOOT)

FORECOOLERS (See WATER COOLERS)

FORGINGS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Aluminum Co. of America, Pittsburgh 19, Pa.
American Brass Co., Waterbury 20, Ct. (*p. 185*)
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Bethlehem Steel Co., Bethlehem, Pa.
Bonney Forge & Tool Wks., Allentown, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Kropp Forge Co., 5301 W. Roosevelt Rd., Chicago 50, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Titan Metal Mfg. Co., Bellefonte, Pa.
Transue & Williams Steel Forging Corp., Alliance, O.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
J. H. Williams & Co., 400 Vulcan St., Buffalo 7, N.Y.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

FOUL GAS EXTRACTORS (See PURGERS)

FRAMES, DISPLAY CASE, etc. (See DISPLAY CASE DOORS)

FREEZERS, BATCH, BLAST, CONTINUOUS, QUICK, SHARP, TUNNEL, etc.

Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Belt-Ice Corp., 2845-16th Ave., W., Seattle 99, Wash. (*p. 126*)
Burge Ice Machine Co., 218 N. Jefferson St., Chicago 6, Ill.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.
Frick Co., Waynesboro, Pa. (*p. 51*)
King Co., 902 N. Cedar St., Owatonna, Minn.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 87*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Votator Div., Girdler Corp., Louisville 1, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

FREEZER DOORS (See COLD STORAGE DOORS)

FREON REFRIGERANTS

Ansul Chemical Co., Marinette, Wis. (Sales Agents) (*p. 115*)
E. I. du Pont de Nemours & Co., Inc., Kinetic Chemicals Div., Wilmington 98, Del. (*p. 118*)

(*Continued*)

✓ *Check these*

OUTSTANDING FEATURES OF

"FREON" SAFE REFRIGERANTS

✓ **Freedom from moisture . . . not more than 10 parts per million in "Freon-12."**
✓ **Narrow boiling-point range . . . confined within limits of 1°C.**
✓ **Not more than 2% non-condensible gases in vapor phase.**
✓ **Freedom from acids. There are none in "Freon."**
✓ **Freedom from high- and low-boiling products.**
✓ **Nontoxic . . . nonflammable . . . nonexplosive.**

Regardless of the temperature required, or the design of the equipment used, there is a "Freon" safe refrigerant that will do the job.

"FREON-11"—CCl_3F—B.P. 74.7°F. (23.7°C.)—For commercial and industrial air conditioning systems employing centrifugal-type compressors.

"FREON-12"—CCl_2F_2—B.P.-21.6°F. (-29.8°C.)—The most widely used of all "Freon" refrigerants. Used in industrial and commercial air conditioning and household refrigerating systems employing rotary or reciprocating-type compressors, and in industrial and commercial low-temperature refrigerating systems employing centrifugal-type compressors.

"FREON-21"—$CHCl_2F$—B.P. 48.0°F. (8.9°C.)—For industrial and commercial air conditioning systems employing centrifugal-type compressors. Also in fractional horsepower refrigerating systems employing rotary-type compressors and in air conditioning systems of the absorption type.

"FREON-22"—$CHClF_2$—B.P.-41.4°F. (-40.8°C.)—Due to its low boiling point, "Freon-22" is ideal for jobs where low temperatures are necessary. May be used in reciprocating or centrifugal-type compressors.

"FREON-113"—$CCl_2F\text{-}CClF_2$—B.P. 117.6°F. (47.6°C.)—Used in air conditioning systems employing centrifugal-type compressors. Not for low-temperature refrigeration.

"FREON-114"—$CClF_2CClF_2$—B.P. 38.4°F. (3.5°C.)—Used as a refrigerant in household systems employing rotary-type compressors. Also in industrial and commercial low-temperature refrigerating systems employing centrifugal-type compressors.

Full details on "Freon" refrigerants are available from E. I. du Pont de Nemours & Co. (Inc.), "Kinetic" Chemicals Division, Wilmington 98, Delaware.

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KINETIC

FREON

SAFE REFRIGERANTS

"Freon" is Du Pont's registered trade-mark for its fluorinated hydrocarbon refrigerants

FREON REFRIGERANTS (Continued)

Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal. (Sales Agents)
Virginia Smelting Co., W. Norfolk, Va. (Sales Agents) (*p. 35*)

FREQUENCY CHANGERS

Forbes & Myers, 173 Union St., Worcester, Mass.

FRICTION TAPE

H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Firestone Industrial Products Co., 1200 Firestone Pkwy. Akron 17, O.
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc, 7800 S. Woodlawn Ave., Chicago 19, Ill.

FROZEN FOOD CABINETS (See HOME & FARM FREEZERS; also DISPLAY CASES, FROZEN FOODS)

FUSES

Bussmann Mfg. Co., University & Jefferson Sts., St. Louis 7, Mo.
Chase-Shawmut Co., Newburyport, Mass.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Jefferson Elec. Co., 25 Ave. & Madison St., Bellwood, Ill.
Littlefuse, Inc., 4757 Ravenswood Ave., Chicago 40, Ill.
Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12, Wis.

FUSIBLE PLUGS

Aminco Refrigeration Products Co., 14544 3rd Ave., Detroit 3, Mich. (*p. 149*)
Chase-Shawmut Co., Newburyport, Mass.
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. (*p. 104*)
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.

GALVANIZING

W. Ames & Co., Jersey City, N.J.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
L. O. Koven & Bro., Inc., 154 Ogden Ave., Jersey City 7, N.J.
Reeves Steel & Mfg. Co., 137 E. Iron Ave., Dover, O.
St. Louis Screw & Bolt Co., 6900 N. B'way, St. Louis 15, Mo.
United Steel & Wire Co., Battle Creek, Mich.

GAS COOLERS

Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Trane Co., La Crosse, Wis. (*p. 118*)
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

GAS MASKS

Acme Protection Equipment Co., 3035 W. Lake St., Chicago 12, Ill.
Central Safety Equip. Co., 2200 E. Huntingdon St., Phila. 25, Pa.
Chicago Eye Shield Co., 2300 Warren Blvd., Chicago 12, Ill.
Davis Emergency Equip. Co., Inc., 45 Halleck St., Newark 4, N.J.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Mine Safety Appliances Co., Braddock, Thomas & Meade Sts., Pittsburgh, Pa.
Pulmosan Safety Equip. Corp., 644 Pacific St., Brooklyn 17, N.Y.

GASES, RARE

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Matheson Co., Inc., P.O. Box 85, E. Rutherford, N.J.

GASKET, DOOR (See DOOR GASKET)

GASKETS & GASKET MATERIAL, PACKING, etc

Anchor Packing Co., 401 N Broad St., Phila. 8, Pa.
Armstrong Cork Co., Lancaster, Pa. (*p. 132*)
Belmont Packing & Rubber Co., Butler & Sepviva Sts., Phila. 7, Pa.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Chamberlin Co. of America, 1254 LaBrosse St., Detroit 26, Mich.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91 Ct.
A. W. Chesterton Co., 4 Ashland St., Everett 49, Mass
Chicago Belting Co., 113 N. Green St., Chicago 7, Ill.
Chicago Gasket Co., 1276 West North Ave., Chicago, Ill.
Chicago Rawhide Mfg. Co., 1306 Elston Ave., Chicago 22, Ill.
Chicago-Wilcox Mfg. Co., 7701 Avalon Ave., Chicago 19, Ill.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Continental Diamond Fibre Co., 3 Chapel St., Newark, Del.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
C. Lee Cook Mfg. Co., 916 S. 8th St., Louisville 3, Ky.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Crane Packing Co., 1800 Cuyler Ave., Chicago 13, Ill.
Dareoid Co., Inc., 145-6th Ave., N.Y.C. 13
Delgro Mfg. Co., Inc., Boonton, N.J.
Detroit Gasket & Mfg. Co., 12640 Burt Rd., Detroit 23, Mich.
Distillation Products Industries, Div. of Eastman Kodak Co., Rochester 3, N.Y.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. (Plastic)
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Endura Mfg. Corp., 45 N. 4th St., Quakertown, Pa.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Excelsior Leather Washer Mfg. Co., Inc., Rockford, Ill. (Leather)
Felters Co., 210 South St., Boston 11, Mass.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Fisher Leather Belting Co., Inc., 325 N. 3rd St., Phila. 6, Pa.
Flexitallic Gasket Co., 934 Eighth St., Camden 2, N.J.
France Packing Co., Tacony, Phila. 35, Pa.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Harper Packing Co., Inc., 312 Patterson St., Chester, Pa.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Hollow Center Packing Co., 6523 Euclid Ave., Cleveland 3, O.
E. F. Houghton & Co.; 303 W. Lehigh Ave., Phila. 33, Pa.

(Continued)

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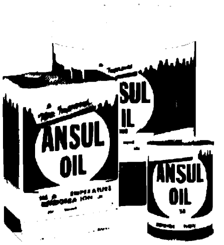

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GASKETS (Continued)

Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Keasbey & Mattison Co., Ambler, Pa.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. *(p. 55)*
Metallo Gasket Co., 10 Bethany St., New Brunswick, N.J.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Quaker Rubber Corp., Tacony & Comly Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
J. E. Rhoads & Sons, 35 N. 6th St., San Francisco 3, Cal.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
Rodpak Mfg. Co., 1315 Natoma St., San Francisco 3, Cal.
Rubatex Div., Great American Industries, Bedford, Va. *(p. 130)*
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
Sponge Rubber Products Co., 1600 Derby Place, Shelton, Ct.
U. S. Graphite Co., Saginaw, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.
Wolverine Fabricating & Mfg. Co., Inc., Princess St. & M.C.R.R., Inkster, Mich.
York Corp., York, Pa. *(p. 52)*

GAUGE GLASSES

Semon Bache & Co., 636 Greenwich St., N.Y.C. 14
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
A. W. Chesterton Co., 4 Ashland St., Everett 49, Mass.
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Corning Glass Wks., Corning, N.Y.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2. Mo.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Fischer & Porter Co., Hatboro, Pa.
Meade Goodloe, 2133 Beechwood Drive, Los Angeles 28, Cal.
Claude S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
Henry Valve Co., Melrose Park, Ill. *(p. 141)*
Paul B. Huyette Co., Inc., 401 N. Broad St., Phila. 8, Pa.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Oil Rite Corp., 3466 S. 13th St., Milwaukee 7, Wis.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Sandee Mfg. Co., 5050 W. Foster Ave., Chicago 30, Ill.
Sarco Co., Inc., 350 Fifth Ave., N.Y.C. 1 *(p. 167)*
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
Shomatic Div., Hartford Machine Screw Co., P.O. Box 1440, Hartford 2, Ct.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 33, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

GAUGE PULSATION ABSORBERS

Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Manning Maxwell & Moore Inc., Elias St., Bridgeport 2, Ct.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Mueller Brass Co., Port Huron, Mich.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES, AIR FILTER

Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
F. W. Dwyer Mfg. Co., 315 S. Western Ave., Chicago 12, Ill.
Claude S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
Hays Corp., Michigan City, Ind.
Manning, Maxwell & Moore, 11 Elias St., Bridgeport 2, Ct.
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 13)*
Paasche Airbrush Co., 1905 Diversey Pkwy., Chicago 14, Ill.
U. S. Gauge, Div of American Machine & Metals, Sellersville, Pa.

GAUGES, AMMONIA, COMPOUND, FREON, PRESSURE, VACUUM, etc.

Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Baker Refrigeration Corp., S. Windham, Me.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Distillation Products Industries, Div. of Eastman Kodak Co., Rochester 3, N.Y.
F. W. Dwyer Mfg. Co., 315 S. Western Ave., Chicago 12, Ill.
General Electric Co., 1 River Rd., Schenectady 5, N.Y
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 183)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
James P. Marsh Corp., Skokie, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Rochester Mfg. Co., Inc., 100 Rockwood St., Rochester 10, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

GAUGES, DRAFT, DUCT PRESSURE, etc.

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
F. W. Dwyer Mfg. Co., 315 S. Western Ave., Chicago 12, Ill
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Hastings Instrument Co., Hampton, Va.
Hays Corp., Michigan City, Ind.
E. Vernon Hill, 6826 W. Highland Ave., Chicago 31, Ill.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.

Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. O. Trerice Co., 1420 W. Lafayette Blvd. Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES, FLOW RATE (See FLOW RATE INDICATORS)

GAUGES, LIQUID LEVEL, TANK, etc. (See also GAUGE GLASSES)

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2 Mich.
Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bijur Lubricating Corp., 151 West Passaic St., Rochelle Park, N.J.
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., P.O. Box 1342, Providence 1, R.I.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 101)*
Ernst Water Column & Gage Co., 250 S. Livingston Ave. Livingston, N.J.
Fischer & Porter Co., Hatboro, Pa.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Frick Co., Waynesboro, Pa. *(p. 51)*
Claud S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
Hays Corp. Michigan City, Ind.
Henry Valve Co., Melrose Park, Ill. *(p. 131)*
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Krueger Sentry Gauge Co., 1056 W. Mason St., Green Bay, Wis.
Liquidepth Indicators, Inc., 42-26-28th St., Long Island City 1, N.Y.
Liquidometer Corp., 41-03-36th St., Long Island City 1, N.Y.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti 14, O.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Rochester Mfg. Co., Inc., 100 Rockwood St., Rochester 10, N.Y.
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 33, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

GAUGES, TEMPERATURE (See THERMOMETERS, DIAL)

GEAR MOTORS (See MOTORS, GEARHEAD)

GEARS

Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Chas. Bond Co., 617 Arch St., Phila., Pa.
Boston Gear Works, Quincy 71, Mass.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Fairfield Mfg. Co., Lafayette, Ind.
W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago 24, Ill.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Ohio Gear Co., 1333 E. 179 St., Cleveland, O.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 43, Pa.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1 N.Y.
Simonds Gear & Mfg. Co., 2501 Liberty Ave., Pittsburgh 22, Pa.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.
Westinghouse Elec. Corp., 200 McCandless Ave., Pittsburgh, Pa.

GENERATORS (See ELECTRIC GENERATING PLANTS; also MOTOR GENERATORS)

GLASS

Semon Bache & Co., 636 Greenwich St., N.Y.C. 14
Corning Glass Works, Corning N.Y.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Hooker Glass & Paint & Mfg. Co., 659 W. Washington Blvd., Chicago, Ill.
Libbey-Owens-Ford Glass Co., Nicholas Bldg., Toledo 3, O.
Mississippi Glass Co., 88 Angelica St., St. Louis 7, Mo.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

GLASS BLOCKS

Hooker Glass & Paint & Mfg. Co., 659 W. Washington Blvd., Chicago, Ill.
Owens-Illinois Glass Co., Duraglas Bldg., Toledo 1, O.
Pittsburgh Corning Corp., 307-4th Ave., Pittsburgh 22, Pa. *(p. 129)*
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5

GLASS FILLERS (See DRINKING WATER COOLER FITTINGS)

GLASS, GAUGE (See GAUGE GLASSES)

GLASS, INSULATING OR HEAT DEFLECTING

Libbey-Owens-Ford Glass Co., Nicholas Bldg., Toledo 3, O.
Mississippi Glass Co., 88 Angelica St., St Louis 7, Mo.
Pittsburgh Corning Corp., 307-4th Ave., Pittsburgh 22, Pa. *(p. 129)*
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Thermopane Div., Libbey-Owens-Ford Glass Co., Nicholas Bldg., Toledo 3, O.

GLASS SPECIALTIES

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Semon Bache & Co., 636 Greenwich St., N.Y.C. 14
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Corning Glass Wks., Corning, N.Y.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Distillation Products Industries, Div. of Eastman Kodak Co., Rochester 3, N.Y.
Fischer & Porter Co., Hatboro, Pa.

GLYCOLS

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Dow Chemical Co., Midland, Mich.
E. I. duPont de Nemours & Co., Inc., Wilmington 98, Del.

GOVERNORS

Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Leslie Co., Lyndhurst, N.J.
Troy Engine & Machine Co., Troy, Pa.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 33, Pa.

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2030 Cameron Avenue, La Crosse, Wisconsin

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Denver, Colo.
Des Moines, Iowa
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Nashville, Tenn.
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New Orleans, La.
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North Tarrytown, N.Y.
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Phoenix, Ariz.
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Portland, Maine
Portland, Ore.
Providence, R.I.
Richmond, Va.
Roanoke, Va.
Rochester. N.Y.
St. Louis, Mo.
St. Paul, Minn.
Saginaw, Mich.
Salt Lake City, Utah
San Antonio, Texas
San Francisco, Calif.
Seattle, Wash.
Sioux City, Iowa
South Bend, Ind.
Spokane, Wash.
Syracuse, N.Y.
Toledo, Ohio
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West Hartford, Conn.
Wichita, Kan.
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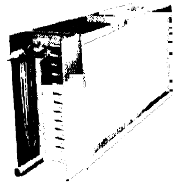
The Trane Company builds a complete line of heating, cooling and air handling equipment. So comprehensive is this line that all major items for any application can be supplied by Trane. This undivided responsibility assures architect, engineer and contractor that each piece of equipment will work with the others to give peak efficiency for the over-all system. From basement equipment room to roof ventilator, it's Trane equipment.

COOLING COILS

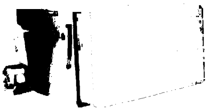
There is a Trane Extended Surface Coil for every comfort cooling or processing application. Trane Cooling Coils come in a wide variety of sizes and arrangements for use with either cold water or direct expansion refrigerants as the cooling medium.



Type DE Cooling Coil

CLIMATE CHANGERS

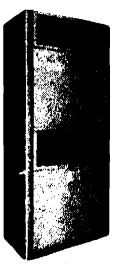
As its name implies, the Climate Changer is a unit type air conditioner for comfort or industrial processing applications. The complete unit can suppply heating, cooling, ventilation, humidification, dehumidification and filtering. It can do all, or any combination, of these jobs. Capacities from 1-60 tons. Also available in multiple zone units.



Horizontal Climate Changer

SELF-CONTAINED AIR CONDITIONERS

For the smaller installation such as home, office, store or shop, these compact units discharge fresh, clean air into the zone of occupancy. Heating coils can be added for year 'round use. These handsomely rugged units are available in sizes from 3 through 10 hp.



Self-Contained Air Conditioner

EVAPORATIVE CONDENSERS

For installations in localities where the use of water for condensing the refrigerant in the air conditioning system is restricted, or where disposal of large quantities of water is a problem. These units condense refrigerants by the evaporative process using a constantly recirculated supply of water. The only water lost is that which is evaporated to cool the water being circulated. Sizes from 3 to 100 tons.

RECIPROCATING COMPRESSORS AND CONDENSER UNITS

Compact, quiet-running Trane Reciprocating Compressors have internal capacity control that adjusts operating level and power needs to cooling loads automatically. Force-feed lubrication, processed valves add years of service to this all-new, all-Trane compressor. Direct-drive, units start unloaded. Six models, 10-50 tons; 4, 6 or 8 cylinders.

TRANE CENTRAVAC

Hermetic centrifugal refrigeration unit—a complete water-chilling system. Completely Trane-designed and built. The CenTraVac features automatic capacity control. No seals or gears. Capacities range from 45-216 tons. Unit performs efficiently at operating levels down to 10% of rated capacity.

TRANE DRY EXPANSION CHILLERS

The Trane Dry Expansion Chiller is designed to provide chilled water for various comfort and process cooling jobs. In this unit the refrigerant is circulated in the tubes of a coil contained within the outer shell. Water is passed over the tubes and chilled. Trane Dry Expansion Chillers have over 3,000 variations in a wide range of capacities.

TRANE AIR WASHERS

Trane Air Washers are of rugged waterproof construction, designed to give lasting trouble-free service. These units come in 5 foot, 7 foot and 9 foot lengths in one stage units. Two stage units are also available. Capacities of Trane Air Washers range from 3,000 to 70,000 cfm.

TRANE CENTRIFUGAL FANS

Recommended for all types of heating, cooling, ventilating and air handling applications. Available in both forward curved and backwardly inclined blade designs, belt or direct drive, single or double widths. Wheel diameters (forward inclined): 4½" to 89"; (backward inclined): 12" to 109".

TRANE EVAPORATIVE COOLERS

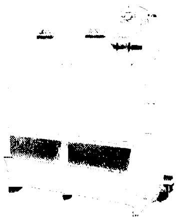
Designed for cooling fluids in a closed system, such as quenching oil, engine jacket water, engine lubricating oil, etc. Using this type of unit eliminates the possibility of contaminating the liquid being cooled and affords cooling without direct contact of fluids with the air.

TRANE PRODUCT COOLERS

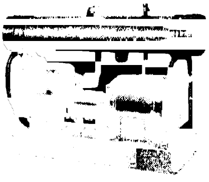
Trane Product Coolers for forced circulation of refrigerated air in walk-in boxes come in 10 sizes with a wide range of capacities.

TRANE CIRCULATING PUMPS

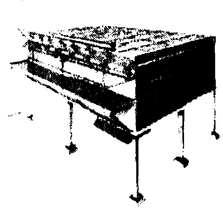
The Trane Type SSU Pump is precision built for mounting directly on the driving motor. Available in five sizes from 1" through 2½" with maximum capacity of 250 gallons per minute against 80 feet, or 125 gallons per minute against a maximum head of 120 feet.



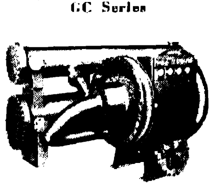
Evaporative Condenser Series 50



Reciprocating Compressor Model 20 CT



Dry Type Fluid Cooler GC Series



Trane CenTraVac

Trane Dry Expansion Chiller

BI Single Width Centrifugal Fan

UNITRANE

Incorporating modern beauty with the latest in air conditioning developments, the revolutionary new UniTrane represents the finest in multi-room air conditioning. UniTrane is ductless, employing a patented moisture controller that is exclusively Trane to give completely independent moisture and temperature control. This feature makes UniTrane the only system that separately processes ventilation air within the unit, entirely eliminating expensive, space-consuming ductwork. Each unit is independent from the other, giving full tenant control and increased flexibility.

TRANE DRY TYPE FLUID COOLER

Condenses refrigerants by air cooling, eliminating water loss and extensive water treatment. Condenses gases in closed system with minimum maintenance. Also suited for cooling engine jacket water and lubricating oil, process cooling and condensing. Available in 22 sizes, with vertical or horizontal air flow.

TRANE AIR CONDITIONING MANUAL

Trane offers the engineering profession an unbiased textbook covering the fundamentals of air conditioning. The Manual not only shows how to design every type of system, but clarifies the underlying principles as well, enabling both student and engineer to reason out their own problems. Price $5.00.

TRANE REFRIGERATION MANUAL

Published primarily as an aid in understanding and correcting installation, maintenance and repair problems. Price $1.50.

OTHER TRANE EQUIPMENT

The complete Trane Line also includes—1. Trane Unit Ventilators for schoolroom air conditioning; 2. Trane Condensation and Centrifugal Pumps for a large variety of uses; 3. Trane Force-Flo Heaters for quiet heat and neat appearance; 4. Trane Railroad and Bus Air Conditioning Equipment of all kinds; 5. Trane Shell and Tube Heat Exchangers for cooling and heating vapors or liquids in a closed system; 6. Transformer Oil Coolers; 7. Convector-radiators; 8. Heating Coils; 9. Unit Heaters—propeller and blower; 10. Roof Ventilators; 11. Steam Heating Specialties; 12. Hot Water Specialties; 13. Wall-Fin Heaters.

GRAPHITE PRODUCTS (See also CARBON PRODUCTS)

Acheson Colloids Corp, Port Huron Mich
Graphite Metallizing Corp, 1050 Nepperhan Ave Yonkers 3 N Y
Speer Carbon Co, St Marys Pa
U S Graphite Co Saginaw Mich

GREASE CUPS (See also LUBRICATORS)

Keystone Lubricating Co 21st & Lippincott Sts Phila 32 Pa
Link Belt Co 220 S Belmont Ave Indpls 6 Ind
Lunkenheimer Co Beekman St & Waverly Ave Cin'ti 14, O
Swift Lubricator Co Inc 101 Home St Elmira N Y

GREASES

Cities Service Oil Co (Pa) 60 Wall Tower N Y C 5
Esso Standard Oil Co 15 W 51st St N Y C 19
Gulf Oil Corp Gulf Bldg Pittsburgh Pa
E F Houghton & Co 303 W Lehigh Ave Phila 33 Pa
Keystone Lubricating Co 21st & Lippincott Sts Phila 32 Pa
Lubriplate Div Fiske Bros Refining Co 129 Lockwood St Newark N J
Macmillan Petroleum Corp 530 W 6th St Los Angeles 14 Cal
Shell Oil Co Inc 50 W 50th St N Y C 20
Standard Oil Co of California 225 Bush St San Francisco 20 Cal
Sun Oil Co 1608 Walnut St Phila 3 Pa

GRILLES (See also AIR DIFFUSERS)

A & A Register Co 8327 Clinton Rd Cleveland 9 O
A J Mfg Co 2119 Washington Kansas City 8 Mo
Air Control Products Inc Coopersville Mich
Anemostat Corp of America, 10 E 39th St, N Y C 16 *p 14*
Auer Register Co 3608 Payne Ave Cleveland O
Bahnson Co 1001 S Marshall St Winston Salem 7 N C
Barber Colman Co Rockford Ill
Best Register Co 2005 W Oklahoma Ave Milwaukee 7 Wis
Central Wire & Iron Wks 1211 E Locust St Des Moines 9 Ia
Controlair Mfg Co 1032 S Compton Ave Los Angeles Cal
Cyclone Fence Div American Steel & Wire Co U S Steel Corp Subsidiary P O Box 260 Waukegan 1 Ill
Diamond Mfg Co Wyoming Pa
Empire Ventilation Equipment Co 35 39 Vernon Blvd Long Island City 1 N Y
General Industries Co Olive & Taylor Sts Elyria O
Harrington & King Perforating Co 5655 Fillmore St Chicago 44 Ill
Hart & Cooley Mfg Co 500 E 8th St Holland Mich
Hendrick Mfg Co Carbondale Pa
Hupp Corp 1250 W 76th St Cleveland 2 O
Independent Register Co 3747 E 93rd St Cleveland 5 O
Lockjoint Wood Products Co 1721 Mildred Ave Wichita 7 Kan
Manhattan Perforated Metal Co Inc 43 17 37th St Long Island City 1 N Y
Minneapolis-Honeywell Regulator Co 2933 4th Ave, S, Minneapolis 8 Minn *p [illegible]*
Newman Bros, Inc 666 W 4th St Cin'ti 3 O
Pyle Natl Co 1371 W 37th St Chicago 9 Ill
St Louis Blow Pipe & Heater Co Inc Div of Skinner Heating & Ventilating Co Inc 1948 N 9th St St Louis 6, Mo
Snyder Equip Co 4115 Edwards Rd Cincinnati 9 O
Amos Thompson Corp Edinburg Ind
E H Titchener & Co 71 Clinton St Binghamton N Y
Titus Mfg Corp Waterloo Ia
Tuttle & Bailey Inc New Britain Ct
U S Air Conditioning Corp Como Ave S E at 33rd St Minneapolis 14 Minn *p 3*
U S Gypsum Co 300 W Adams St Chicago 6 Ill
U S Register Co 314 E Burnham St Battle Creek Mich
United Steel & Wire Co Battle Creek Mich
Wall Wire Products Co Plymouth Mich
Young Regulator Co 5203 Euclid Ave Cleveland O

GROMMETS (See WASHERS also RUBBER PRODUCTS, MOLDED)

GUARDS, FAN, MACHINERY etc (See also WIRE FORMING)

Buffalo Wire Wks 150 Terrace Buffalo 2 N Y
Central Wire & Iron Wks 1211 E Locust St Des Moines 9 Ia
Collis Co Box 231 Clinton Ia *p [illegible]*
Cyclone Fence Div American Steel & Wire Co U S Steel Corp Subsidiary P O Box 260 Waukegan 1 Ill
Dahlstrom Metallic Door Co 43[illegible] Buffalo St Jamestown N Y
[illegible]
Harrington & King Perforating Co 5655 Fillmore St Chicago 44 Ill
Kentucky Metal Products Co Preston St & Audubon Park Louisville 4 Ky
Pulmosan Safety Equip Corp [illegible] St Brooklyn 17 N Y
Edw Renneburg & Sons Co [illegible] Boston St Baltimore 24 Md
St Louis Blow Pipe & Heater Co Inc Div of Skinner Heating & Ventilating Co Inc 1948 N 9th St St Louis 6 Mo
E H Titchener & Co 71 Clinton St Binghamton N Y
U S Gypsum Co 300 W Adams St Chicago 6 Ill
Union Steel Products Co 448 Pine St Albion Mich *p [illegible]*
United Steel & Wire Co Battle Creek Mich
United Welding Div General Machinery Corp 545 N 3rd St Hamilton O
Wall Wire Products Co Plymouth Mich

HAIR FELT (See FELT also INSULATION HAIR

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Capitol Mfg. & Supply Co., 153 W Fulton St., Columbus, O.
Chicago Expansion Bolt Co., 1338 W Concord Place, Chicago 22, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (p 109)
Frick Co., Waynesboro, Pa. (p 51)
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R I
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill
Goodloe E. Moore Co., 2811 Vermilion, Danville, Ill
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
M. B Skinner & Co , South Bend 23, Ind
E H Titchener & Co , 76 Clinton St , Binghamton, N Y
Walworth Co., 60 E. 42nd St , N.Y C 17

HARDENING ROOMS (See ICE CREAM HARDENING ROOMS)

HARDWARE, COLD STORAGE DOOR

Arcade Mfg. Div., Rockwell Mfg. Co., 1212 E Shawnee St., Freeport, Ill.
Butcher Boy Cold Storage Door Co., 170 N. Sangamon St., Chicago 7, Ill. (p 45)
Chase Industrial Refrigerator Equip & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
Cork Insulation Co , Inc , 155 E 44th St., N.Y.C 17
Grand Rapids Brass Co , 60 Scribner Ave , Grand Rapids, Mich
Jamison Cold Storage Door Co , Hagerstown, Md
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.
Koch Butchers' Supply Co , 600 E. 14th Ave., N Kansas City, Mo.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 (p 136)
V E Sprouse Co , Inc , Columbus, Ind
Standard-Keil Hardware Mfg. Co , Inc., 639 Broadway, N Y C 12
Streator Products Corp., 508 N. 8th St , Fairfield, Ia.
York Corp., York, Pa. (p [illegible])

HARDWARE, REFRIGERATOR

Arcade Mfg. Div., Rockwell Mfg. Co , 1212 E Shawnee St., Freeport, Ill.
Chase Industrial Refrigerator Equip & Engrg. Co., 630 Reading Rd., Reading, O.
Corbin Cabinet Co , Div of American Hardware Corp , New Britain, Ct
Croname, Inc , 3701 N Ravenswood Chicago 13, Ill
Dent Hardware Co , 1102 3rd St , Fullerton Pa
Grand Rapids Brass Co , 60 Scribner Ave , N W , Grand Rapids 1, Mich.
Kason Hardware Corp , 127 Wallabout St , Brooklyn 6, N.Y.
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Koch Butchers' Supply Co , 600 E. 14th Ave , N Kansas City 16, Mo.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
National Lock Co , 7th St & 18th Ave , Rockford Ill
V. E Sprouse Co , Inc , Columbus, Ind
Standard-Keil Hardware Mfg. Co., Inc., 639 Broadway, N.Y C. 12
Streator Products Corp , 408 N. Depot St , Fairfield Ia

HARDWARE SPECIALTIES (See also certain specific items)

Arcade Mfg. Div., Rockwell Mfg. Co., 1212 E. Shawnee St., Freeport, Ill
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill
Fox Co., Fox Lane, Cin'ti. 23, O.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co , 60 Scribner Ave , N.W , Grand Rapids 1, Mich.
Kason Hardware Corp., 127 Wallabout St , Brooklyn 6, N.Y.
Koch Butchers' Supply Co., 600 E. 14th Ave., N Kansas City 16, Mo.
National Lock Co , 7th St. & 18th Ave , Rockford, Ill
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
V. E. Sprouse Co , Inc , Columbus, Ind
Standard-Keil Hardware Mfg. Co., Inc , 639 Broadway, N.Y.C 12
Stronghold Screw Products, Inc , 216 W. Hubbard St., Chicago 10, Ill.
Swift Mfg. Co., Inc , 1455 E. Nine Mile Rd , Hazel Park, Mich.
Townsend Co , 205 River Rd , New Brighton, Pa
L R Wagner Mfg Co 4001 N 32nd St , Milwaukee, Wis

HARNESSES, ELECTRICAL (See WIRING DEVICES)

HEADS, TANK, etc. (See TANK HEADS)

HEAT EXCHANGERS

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (p. 211)
Aerofin Corp., 410 Geddes St., Syracuse, N.Y. (p 11)
Richard M Armstrong Co , Box 188, W Chester Pa
Baker Refrigeration Corp., S. Windham, Me. (p. 63)
Bell & Gossett Co., Morton Grove, Ill. (p 209)
Betz Corp , 445 State St., Hammond, Ind
Binks Mfg. Co., 3124-32 Carroll Ave , Chicago 16, Ill. (p 28)
Bohn Aluminum & Brass Corp , E. Maumee, Adrian, Mich.
Cherry-Burrell Corp , 427 W. Randolph St , Chicago 6, Ill

(Continued)

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HEAT EXCHANGERS (Continued)

Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. *(p. 152)*
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Eastern Industries, Inc., 296 Elm St., New Haven, Ct.
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Franklin Body & Equip. Corp., 1042 Dean St., Brooklyn 16, N.Y.
Frick Co., Waynesboro, Pa. *(p. 51)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Kirsch Co., Sturgis, Mich. (Hydrogen brazed)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 153)*
Kramer-Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 195)*
Larkin Coils, 519 Memorial Dr , S.E., Atlanta 1, Ga. *(p. 41)*
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
Lummus Co., 420 Lexington Ave., N.Y.C.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 75)*
Mueller Brass Co., Port Huron, Mich
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
J. F. Pritchard & Co., 908 Grand Ave., Kansas City 6, Mo. *(p. 72)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Remco, Inc., Zelienople, Pa. *(p. 86)*
Edw. Henneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 105)*
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. *(p. 23)*
Super-Cold Corp., 1020 E. 59th St., Los Angeles, Cal.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 121)*
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p. 89)*
Trane Co., La Crosse, Wis. *(p. 118)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Votator Div., Girdler Corp., Louisville 1, Ky. (Mechanical)
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wittenmeier Machinery Co., 850 N Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J *(p. 144)*
York Corp., York, Pa. *(p. 52)*
Young Radiator Co., Racine, Wis. *(p. 77)*

HEAT EXCHANGER TEES

Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.

HEAT PUMPS (See CONDENSING UNITS)

HEAT TREATING

Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.J.
Paulo Products Co., 5711 W. Park Ave., St. Louis 10, Mo

HEATERS, & HEATING ELEMENTS, ELECTRIC

American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Carborundum Co., P.O. Box 337, Niagara Falls, N.Y.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Electric Heater Co., Woodend Rd., Stratford, Ct.
Electromode Corp., 45 Crouch St., Rochester 3, N.Y.
Falcon Equip. Co., 13th St., New Brighton, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.
Westinghouse Elec. Corp., Meadville, Pa.
Westinghouse Elec. Corp., P.O. Box 7348, S. Philadelphia 1, Pa.
Edwin L. Wiegand Co., 7506 Thomas Blvd., Pittsburgh 8, Pa.

HEATERS, AIR, GAS FIRED

Acme Equip. Co., 205 E. Broadway, Muskogee, Okla.
Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.
Central States Mfg. Co., Inc., 1200 S. Summit, Arkansas City, Kan.
Trane Co., La Crosse, Wis. *(p. 118)*

HEATING COILS (See AIR CONDITIONING COILS

HERMETIC COMPRESSORS (See CONDENSING UNITS)

HERMETIC HOUSINGS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 178)*
Blissfield Mfg. Co , 626 Depot St., Blissfield, Mich.

HERMETIC REFRIGERATING SYSTEMS

Tecumseh Products Co., Tecumseh, Mich. *(p. 61)*
Trane Co., La Crosse, Wis. *(p. 118)*
York Corp., York, Pa. *(p. 52)*

HERMETIC TERMINALS (See TERMINALS, HERMETIC)

HIGHSIDE FLOATS (See also FLOAT SWITCHES)

(A—Ammonia; B—Other refrigerants)

(A,B) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
(B) Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 149)*
(A,B) Armstrong Machine Wks., 831 Maple St., Three Rivers, Mich. *(p. 160)*
Crider Corp., 2216 Northland, Lakewood, O.
(A,B) Frick Co., Waynesboro, Pa. *(p. 51)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
(A,B) Magnetrol, Inc., 2110 S. Marshall Blvd., Chicago 23, Ill.
(A,B) H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. (p. *89*)
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.
(A) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A,B) York Corp., York, Pa. *(p. 52)*

HINGES, REFRIGERATOR (See HARDWARE)

HOISTS (See CRANES)

HOME & FARM FREEZERS; FROZEN FOOD CABINETS (See also DISPLAY CASES, FROZEN FOOD)

Amana Refrigeration, Inc., Amana, Ia.
American Refrigeration Corp., 1025 E. Excelsior Ave., Hopkins, Minn.
Artkraft Mfg. Corp., Kibby St. & D.T.&I.R.R., Lima, O.
Aug. G. Barkow Mfg. Co., 2220 S. 43rd St., Milwaukee, Wis.
Ben-Hur Mfg. Co., 634 E. Keefe Ave., Milwaukee 12, Wis.
Brewer-Titchener Corp., Binghampton, N.Y.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p, 25)*
Coolerator Co., 50 Ave W., and Wadena St., Duluth 1, Minn.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y.
Crosley Div., Avco Corp., Cin'ti. 25, O.
Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
Deepfreeze Div., Motor Products Corp., 2301 Davis St., N. Chicago, Ill.
Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
Esco Cabinet Co., West Chester, Pa.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Franklin Transformer Co., 65-22nd., N.E., Minneapolis 18, Minn.
Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 3, Tex.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Gibson Refrigerator Co., 515 W. Williams St., Greenville, Mich.
Harder Refrigerator Corp., Div. of Tyler Fixture Corp., Cobleskill, N.Y.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
Holverson Co., 209 E. Lake St., Minneapolis 3, Minn.
Hotpoint, Inc., 5600 W. Taylor St., Chicago 44, Ill.
Hupp Corp., 1250 W. 76th St., Cleveland 2, O.
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p 83)*
Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6. Mo.
International Harvester Co., 180 N. Michigan Ave, Chicago 1, Ill.
Jordon Refrigerator Co., 58th St. & Grays Ave., Philadelphia 43, Pa.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit Mich. *(p. 21)*
Jack Langston Co., 3709 Elm St., Dallas, Tex.
Leonard Div., Nash-Kelvinator Corp , 14250 Plymouth Rd., Detroit 32, Mich.
McCall Refrigerator Corp., Hudson, N.Y.
McCray Refrigerator Co., Kendallville, Ind.
Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
Masterfreeze Corp., Sister Bay, Wis.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nelson Mfg. Co., 4016 N. Union St., St. Louis 15, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1. Md.
Philco Corp., Tioga & C Sts., Phila. 34, Pa.
Portable Elevator Mfg. Co., 920 E. Grove St., Bloomington, Ill.
Revco, Inc., Deerfield, Mich.
Sanitary Refrigerator Co., Fond du Lac, Wis.
Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
Simplex Mfg. Co., 1135-3rd St., Oakland 20, Cal.
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Stoddard Mfg. Co., 617-4th St., S.W., Mason City, Ia.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
United Refrigerator Co., Hudson, Wis.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
Westinghouse Elec. Corp., 653 Page Blvd., Springfield 2, Mass.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.
Yoder Corp., 133 W. 80th St., Cleveland 2, O.
York Corp., York, Pa. *(p. 52)*

HOOKS (See HARDWARE SPECIALTIES)

HOSE, FLEXIBLE METAL (See also CHARGING LINES; also FLEXIBLE CONNECTIONS; also TUBING, FLEXIBLE)

Aeroquip Corp , 300 S. East Ave., Jackson, Mich.
Allied Metal Hose, 37-46 9th St., Long Island City, N.Y.
American Brass Co., Waterbury 83, Ct. *(p. 189)*
Atlantic Metal Hose Co., Inc., 124 W. 64th St., N.Y.C. 23
Chicago Metal Hose Corp., Maywood, Ill.
Clifford Mfg Co. Grove St., Waltham, Mass.
Dole Refrigerating Co. 5910 N. Pulaski Rd., Chicago 30 Ill.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill
Flexible Tubing Corp., New Whitfield St., Guilford, Ct.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Johnson Metal Hose Co., 226 Mill St., Waterbury 88, Ct.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y.
Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
Universal Metal Hose Co., 2125 S. Kedzie Ave., Chicago, Ill.

HOSE, RADIATOR, WATER, AIR, etc.

Boston Woven Hose & Rubber, P.O. Box 1071, Boston 3, Mass.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Flexible Tubing Corp., New Whitfield St., Guilford, Ct.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Quaker Rubber Corp., Tacony & Comly Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Thermoid Co., Trenton, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

HOSE CLAMPS (See FITTINGS, HOSE)

HUMIDIFIERS & HUMIDIFYING SYSTEMS (See also SPRAY HUMIDIFIERS)

Abbeon Supply Co., 58-10-41st Drive, Woodside, N.Y.C.
Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Coils Co., 25 Lexington St., Newark 5, N.J.
American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
American Moistening Co., Div. of Grinnell Corp., 260 W. Exchange St., Providence 1, R.I.
Anetsberger Bros., Northbrook, Ill.
Armstrong Machine Wks., 831 Maple St., Three Rivers, Mich. *(p. 160)*
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Binks Mfg. Co., 3124-32 Carroll Ave., Chicago 16, Ill. *(p. 73)*

(Continued)

HUMIDIFIERS (Continued)

Bowser Technical Refrigeration, Div of Bowser, Inc, Terryville, Ct
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Drying Systems, Inc., 1810½ Foster Ave., Chicago 40, Ill.
Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal
Grinnell Co., Inc., 260 W. Exchange St, Providence 1, R.I.
Kauffman Air Conditioning Corp, 4336 W Pine Blvd, St Louis, Mo
McDonnell & Miller, Inc, 3500 N Spaulding Ave, Chicago 18, Ill
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd, Chicago 41, Ill
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. [illegible])*
D. J. Murray Mfg. Co, 1024 3rd St, Wausau Wis
National Air Conditioning, Inc, Johnstown, Pa
National Engrg & Mfg Co., 519 Wyandotte, Kansas City 8, Mo.
Paasche Airbrush Co, 1909 Diversey Pkwy Chicago 14, Ill
Parks-Cramer Co., Box 444, Fitchburg, Mass
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O
Rhode Island Humidifier & Ventilating Co, 99 Chauncey St., Boston 11, Mass.
St. Louis Blow Pipe & Heater Co., Inc., Div of Skinner Heating & Ventilating Co., Inc., 1918 N 9th St., St. Louis 6, Mo.
H. J. Somers, Inc., 6063 Wabash Ave, Detroit 8, Mich
Spray Engrg Co, 114 Central St Somerville 45 Mass
Spraying Systems Co, 3201 Randolph St, Bellwood, Chicago, Ill
B. F. Sturtevant Div. Westinghouse Elec. Corp., 101 Readville St., Boston 39, Mass. *(p. [illegible]3)*
Supreme Elec. Products Co., 194 Vassar St, Rochester 7 N.Y.
Surface Combustion Corp 2375 Dorr St Toledo 1, O
Techtman Industries, 714 W Wisconsin Ave, Milwaukee 1, Wis.
Trane Co., La Crosse, Wis. *(p. 118)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd, Minneapolis 14, N.J. *(p. 37)*
Viking Air Conditioning Corp, 5600 Walworth Ave, Cleveland 2, O.
Walton Laboratories, Inc, 1186 Grove St, Irvington 11 N J
Water Cooling Corp., 71 Nassau St., N.Y C. 7
Stanley Wertheim & Son, 58 01-41st Dr., Woodside, N.Y.C.
York Corp., York, Pa. *(p. [illegible]2)*

HUMIDISTATS (See CONTROLS, HUMIDITY)

HYDROGEN

Air Reduction Sales Co, 60 E 42nd St, N.Y C. 17
Linde Air Products Co, Unit of Union Carbide & Carbon Corp, 30 E. 42nd St., N Y C. 17

ICE CANS

Frick Co., Waynesboro, Pa. *(p. [illegible])*
Knickerbocker Stamping Co, Parkersburg, W Va
Jos A Martocello & Co, 229 N 13th St, Philadelphia 7, Pa
Maysteel Products, Inc., 740 N. Plankinton Ave, Milwaukee 3, Wis
Reco Products Div, Refrigeration Engrg Corp, 2020 Naudain St, Phila. 46, Pa
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 5[illegible])*

ICE CREAM CABINETS

Ace Cabinet Corp., New Bedford, Mass.
American Refrigeration Corp, 1025 E Excelsior Ave, Hopkins, Minn
Anheuser Busch, Inc., 9th & Wyoming, St. Louis, Mo
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Brewer-Titchener Corp, Binghamton, N Y.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St, Binghamton, N Y.
Evans Mfg Corp., 460 S. 10th Ave, Mt. Vernon, N Y.
Federal Refrigerator Mfg Co, 550 Elizabeth St, Waukesha, Wis
Fogel Refrigerator Co 5400 Eadom St, Phila, Pa
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 1[illegible])*
C V Hill & Co Inc, 360 Pennington Ave Trenton 1 N J
Jordon Refrigerator Co, 58th & Grays Sts, Philadelphia 8, Pa
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Stanley Knight Corp, 3130 N. Pulaski Rd, Chicago 41 Ill
Liquid Carbonic Corp, 3100 S. Kedzie Ave., Chicago 23, Ill.
Maysteel Products, Inc, 740 N. Plankinton Ave, Milwaukee 3, Wis.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 58)*
Nelson Mfg. Co, 4016 N. Union St, St Louis 15, Mo
Pennco, Inc, 201 N Broad St., Phila. 7, Pa
Refrigeration Corp of America, Div of Lonergan Mfg Co, Albion, Mich.
Savage Arms Corp, 311 Turner St., Utica 1, N Y
Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
Sweden Freezer Mfg Co 3401 17th Ave, W Seattle 99, Wash.
Emery Thompson Machine & Supply Co, 1349 Inwood Ave, N.Y C 52
Tuthill Pump Co 939 E 95th St Chicago 19, Ill
Victor Products Corp Hagerstown, Md
Weber Showcase & Fixture Co, Inc, P.O Box 2018, Los Angeles 54, Cal

ICE CREAM FREEZERS

Bastian-Blessing Co, 4201 W Peterson Ave, Chicago 40, Ill
Cherry-Burrell Corp, 127 W. Randolph St, Chicago 6 Ill.
Creamery Package Mfg. Co, 1243 W. Washington Blvd., Chicago 7, Ill. *(p. [illegible])*
Evans Mfg Corp, 460 S 10th Ave Mt Vernon N Y
C V Hill & Co, 360 Pennington Ave Trenton N J
Maysteel Products, Inc 740 N Plankinton Ave, Milwaukee 3, Wis
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. [illegible])*
Sweden Freezer Mfg Co, 3401 17th Ave, W Seattle 99 Wash
Tekni-Craft, Rockton, Ill
Emery Thompson Machine & Supply Co, 1349 Inwood Ave., N.Y C. 52
Tuthill Pump Co, 939 E 95th St, Chicago 19, Ill
Votator Div, Girdler Corp, Louisville 1, Ky.
Wilson Refrigeration Inc Smyrna Del

ICE CREAM HARDENING CABINETS

Bastian-Blessing Co, 4201 W. Peterson Ave, Chicago 40 Ill.
Brewer Titchener Corp, Binghamton N Y
Evans Mfg Corp, 460 S. 10th Ave., Mt Vernon, N Y
Federal Refrigerator Mfg Co, 550 Elizabeth St, Waukesha Wis
Holverson Co, 209 E. Lake St, Minneapolis 3, Minn.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 58)*
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo (Sectional Construction)
Nelson Mfg. Co, 4016 N. Union St., St. Louis 15, Mo
Reco Products Div, Refrigeration Engrg. Corp., 2020 Naudain St, Phila. 46, Pa.
Schaefer, Inc., 801 Washington Ave., N, Minneapolis 1, Minn.
Super-Cold Corp, 1020 E 59th St., Los Angeles 1, Cal
Sweden Freezer Mfg Co., 3401-17th Ave W, Seattle 99 Wash.
Tekni-Craft, Rockton, Ill
Wilson Refrigeration, Inc., Smyrna Del.
York Corp., York, Pa. *(p. [illegible])*

ICE CREAM HARDENING ROOMS

Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
John Herrel & Sons, Inc., 244 Lear St., Columbus 6, O.
King Co., 902 N. Cedar St., Owatonna, Minn.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo. (Sectional Construction)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Sweden Freezer Mfg. Co., 3401-17th Ave., W., Seattle 99, Wash.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*

ICE CUBE CUTTERS

Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
Gifford-Wood Co., Hill St., Hudson, N.Y.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Thermo Cuber Co., Inc., 2120 N. Southport Ave., Chicago 14, Ill.

ICE CUBE FREEZING LOWSIDES

(A—Ammonia; B—Other refrigerants)

(B) American Refrigeration Corp., 1025 E. Excelsior Ave., Hopkins, Minn.
(B) **Baker Refrigeration Corp., S. Windham, Me.** *(p. 63)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(B) **Bush Mfg. Co., 179 South St., W. Hartford 10, Ct.** *(p. 40)*
(A,B) **Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y.** *(p. 25)*
(B) **Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O.** *(p. 17)*
(B) **Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J.** *(p. 195)*
(A,B) **McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn.** *(p. 199)*
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B) **Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal.** *(p. 106)*
(B) Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Super-cold Corp., 1020 E. 59th St., Los Angeles, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B) **York Corp., York, Pa.** *(p. 52)*

ICE CUBE MACHINES, SELF-CONTAINED

Ajax Corp., Evansville, Ind
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Ice-Flo Corp., Mill St., Providence, R.I.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
Mills Industries, Inc., 4100 Fullerton Ave., Chicago 39, Ill. *(p. 58)*
Nor-Lake, Inc., 2nd & Elm Sts., Hudson, Wis.
United Refrigerator Co., Hudson, Wis.
York Corp., York, Pa. *(p. 52)*

ICE CUBE TRAYS

Aluminum Goods Mfg. Co., Manitowoc, Wis.
Hoosier Cardinal Corp., 601 E. Eichel Ave., Evansville 7, Ind.
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Plastray Corp., 823 Fisher Bldg., Detroit 2, Mich.
Premium Plastics, Inc., 627 Salem Ave., Dayton 6, O.
Harry Zysman, Inc., 427 E. 146th St., N.Y.C. 55 *(p. 125)*

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ICE MAKERS, BULK

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (p 25)
Esco Cabinet Co., West Chester, Pa
Frick Co., Waynesboro, Pa. (p 61)
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (p. 17)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (p 39)
La Crosse Cooler Co , 2800 Losey Blvd , S , La Crosse, Wis.
Nor-Lake, Inc , 2nd & Elm Sts Hudson, Wis
R Perlick Brass Co , 3110 W. Meinecke Ave , Milwaukee 10, Wis
Reco Products Div , Refrigeration Engrg Corp , 2020 Naudain St., Phila 46, Pa
Tenney Engrg , Inc , 26 Ave B, Newark 5 N J
Victor Products Corp , 901 Pope Ave , Hagerstown, Md.
Wilson Refrigeration, Inc , Div of Wilson Cabinet Co , Inc , Smyrna Del

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ICE MAKING MACHINES, AUTOMATIC (See also ICE CUBE MAKING MACHINES)

Baker Refrigeration Corp., S Windham, Me. (p 63)
Belt-Ice Corp., 2845-16th Ave , W., Seattle 99, Wash (p 126)
Flakice Corp , 360 Furman St , Brooklyn 2, N.Y.
Gay Engrg Co., 2730 E. 11th St , Los Angeles 23, Cal.
Kohlenberger Engrg Corp., 1600 W. Commonwealth, Fullerton, Cal. (p 39)
Nor-Lake, Inc , 2nd & Elm Sts , Hudson, Wis
Orley Freezers, Inc., 680 E Fort St , Detroit 26, Mich
Reco Products Div , Refrigeration Engrg Corp , 2020 Naudain St , Phila 46, Pa
United Refrigerator Co , Hudson, Wis
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (p 33)
Henry Vogt Machine Co , 10th & Ormsby St , Louisville 10, Ky
Worthington Pump & Machinery Corp., Harrison, N.J. (p 144)
XL Refrigerating Co , 1831 W 59th St , Chicago 36, Ill
York Corp., York, Pa. (p 12)

INDICATORS (See particular type, i e., FLOW RATE; LIQUID; LEAK; MOISTURE, etc.)

INHIBITORS, CORROSION (See WATER TREATING)

INJECTORS, REFRIGERANT

Cold Control, Inc , 111 Broadway N Y C 6
H. A. Phillips & Co., 3255 W Carroll Ave., Chicago 24, Ill. (p 126)

INSERTS FOR MOLDING, etc (See FASTENERS; also HARDWARE SPECIALTIES)

INSTRUMENT PANELS

Anderson & Sons, Inc N Elm St , Westfield, Mass
Baker Refrigeration Co , S. Windham, Me. (p 63)

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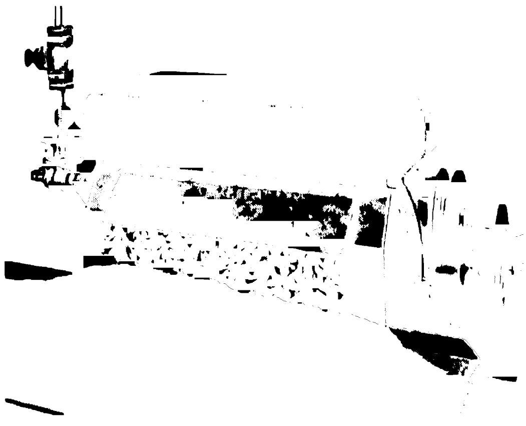

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Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Fischer & Porter Co., Hatboro, Pa.
Claude S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
James P. Marsh Corp., Skokie, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
Panellit, Inc., 7212 North Clark St., Chicago 26, Ill.
Amos Thompson Corp., Edinburg, Ind.

INSULATING PLASTER & CONCRETE

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Forty-Eight Insulations, Inc., Aurora, Ill.
Great Lakes Carbon Corp., 18 E. 48th St., New York 17, N.Y.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robert A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 *(p. 136)*
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5
Western Mineral Products Co., 1720 Madison St., N.E., Minneapolis 13, Minn.
Zonolite Co., 135 S. La Salle St., Chicago 3, Ill.

INSULATION, AEROGEL

Merrimac Div., Monsanto Chemical Co., Everett 49, Mass.

INSULATION, ASBESTOS, MAGNESIA, etc.

Acme Asbestos Covering & Flooring Co., 236 N. Elizabeth St., Chicago 7, Ill.
Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 130 W. 19th St., N.Y.C. 11
Keasbey & Mattison Co., Ambler, Pa.
Mundet Cork Corp., 7101 Tonnelle Ave., N. Bergen, N.J. *(p. 138)*
Norristown Magnesia & Asbestos Co., Washington St., below Ford St., Norristown, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Standard Asbestos Mfg. Co., 860 W. Evergreen Ave., Chicago 22, Ill.
Standard Asbestos Mfg. & Insulation Co., 410 N. Olive St., Kansas City 1, Mo.
Taft-Jenkins Co., Inc., 27 Sargeant St., Holyoke, Mass.
Grant Wilson, Inc., 315 S. Sherman Ave., Chicago 4, Ill.

INSULATION, BALSA

Balsa Ecuador Lumber Corp., 500-5th Ave., N.Y.C. 18
Geo. D. Emery Co., 220-11th Ave., N.Y.C. 1
International Balsa Corp., 96 Boyd Ave., Jersey City 4, N.J.

INSULATION, CELLULAR RUBBER

B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
C. T. Hogan & Co., Inc., 383 Madison Ave., N.Y.C. 17
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 85)*
Rubatex Div., Great American Industries, Inc., Bedford, Va. *(p. 130)*
Sponge Rubber Products Co., 1600 Derby Pl., Shelton, Ct.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

INSULATION, CELLULOSE FIBRE

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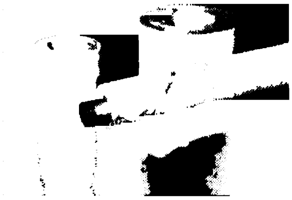

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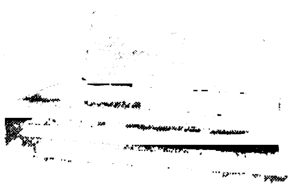

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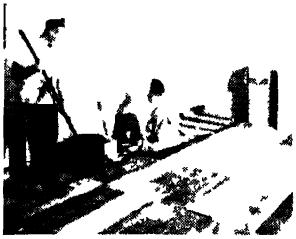

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OWENS-CORNING FIBERGLAS CORPORATION

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Offices Atlanta Baltimore Boston Buffalo Chicago Cincinnati Cleveland, Columbus Dallas Dayton Denver Detroit Grand Rapids Houston Kansas City Louisville Los Angeles Milwaukee Minneapolis New York Philadelphia Pittsburgh Portland Richmond St Louis San Francisco Seattle, Toledo Tulsa Washington D C

FIBERGLAS IS IN YOUR LIFE...FOR GOOD!

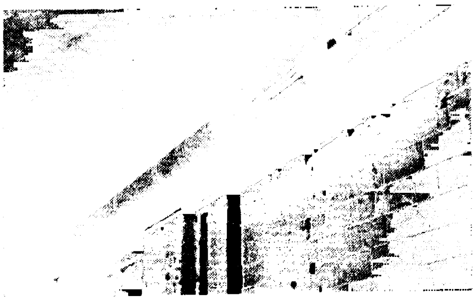
Pittsburgh Corning Corporation

Makers of PC Foamglas Insulation

Dept. AB-01, 307 Fourth Avenue Pittsburgh 22, Pa.

Control Cooling Costs with PC FOAMGLAS ®

PC Foamglas Insulation is used on the walls and ceiling of this cooling room—designed for 32°F. temperature—and on cold pipe lines at the new canning plant of the Krey Packing Company, Belleville, Ill. Insulation Contractor: Gordon Insulation Company, E. St. Louis, Ill.



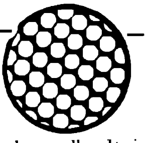
The wide acceptance of PC Foamglas—evidenced by a long list of prominent users—is due mainly to two facts. It is an effective insulation. It is an economical insulation.

The insulating effectiveness of Foamglas helps maintain low temperatures with fewer hours of compressor operation, thus saving money on electric current and postponing costly repairs to cooling equipment.

The cellular glass structure that makes Foamglas such an exceptionally effective insulation also makes it the long life insulation. Since glass is practically impervious to elements that destroy other materials, PC Foamglas—when properly installed—retains its original insulating effectiveness.

Foamglas is giving many years of trouble free service in walls and floors, roofs and ceilings, on piping and fittings, indoors and outdoors, on tanks and processing equipment. Its freedom from costly maintenance, repairs and replacement is another important factor in keeping insulating costs down to rock bottom.

When you are facing special insulating problems, our specialists will be glad to consult with you. Meanwhile, you will find a lot of helpful information in our current literature. Just drop us a line and we shall be glad to send you our booklets and a sample of PC **Foamglas.**



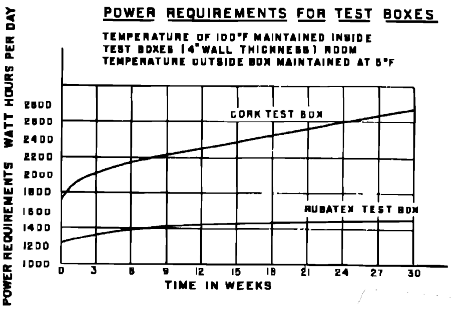
The *best* glass insulation is cellular glass. The *only* cellular glass insulation is PC Foamglas. This unique material is composed of still air, sealed in minute glass cells. It is light-weight, incombustible, verminproof. It has unusually high resistance to moisture, chemicals and many other elements that cause insulation to deteriorate.

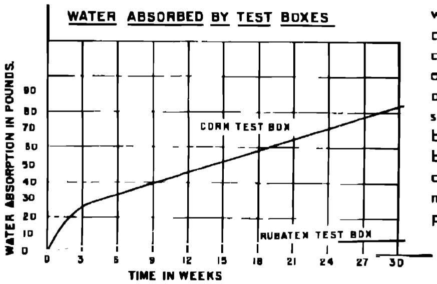
® PITTSBURGH PC CORNING **FOAMGLAS** cellular glass **INSULATION**

. . . when you insulate with Foamglas . . . the insulation lasts!

For lowest power cost—use

RUBATEX INSULATION HARDBOARD





These two graphs show clearly why oversize compressors are customarily installed to overcome later inefficiencies of ordinary insulations which absorb moisture. They show too, that the stability of RUBATEX Hardboard means lower operating cost with more economical power installation.

Power economy, plus an initially lower K-factor of 0.21, plus freedom from maintenance are reasons why more and more freezer plants, cold storage rooms, refrigerated railway cars and trucks are insulated with RUBATEX Insulation Hardboard.

RUBATEX Hardboard has zero moisture absorption. No added vapor barrier is needed. It has good structural strength—weighs only 1½ pounds per cubic foot and will not settle or crumble.

When you insulate, plan for lowest power cost—use RUBATEX Insulation Hardboard. Write for Design Data Book RBH-1-50, Great American Industries, Inc., RUBATEX DIVISION, BEDFORD, VIRGINIA.

Closed Cells shut out heat, cold and moisture. Photo micrograph of a section of RUBATEX Insulation Hardboard shows the dense structure of individually closed cells which give this product its exceptional insulating properties.

RUBATEX INSULATION HARDBOARD

"K" FACTOR 0.21 • ZERO MOISTURE ABSORPTION • NO ADDED VAPOR BARRIER

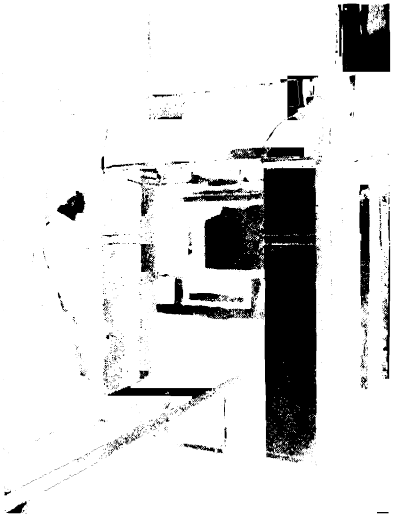
Wood Conversion Company

Dept. 200-1, First National Bank Building, St. Paul 1, Minn.

New York
Boston
Denver
Detroit
Chicago
Kansas City

K-25 INSULATION FIBER*

PNEUMATIC SYSTEM FOR REFRIGERATOR CABINETS



Installation

K-25 Fiber*, applied by the Wood Conversion Company's high-speed pneumatic system, makes it possible to insulate sealed type refrigerator cabinets and doors in a few seconds, with positive automatic control and elimination of all joints, laminations and voids.

K-25 Fiber was scientifically developed by Wood Conversion Company to meet the need for streamlined production of domestic refrigerators. Shipped in highly compressed, easily stored bales, K-25 is quickly processed to the proper low density in your own plant then blown into cabinets and doors under high pressure. Made of clean, new wood fiber, K-25 has an unusually low "k" factor. It is moisture resistant, clean and odorless. Because of its high felting qualities and resilience, K-25 Fiber will not settle.

CABINETS AUTOMATICALLY INSULATED

In the K-25 Fiber Pneumatic System, the refrigerator is automatically insulated under pressure with the insulating fiber conforming to the contour of the outer shell of the cabinet. In the blowing operation, a dummy liner or fixture is used. The fiber completely fills the space between liner and outer shell. In other words, the Pneumatic System introduces the bulk insulating fiber into the cabinet or door and builds or "casts" a self-sustaining mat without the use of any binder. This is made possible by the characteristics of the K-25 Fiber, which are its ability to felt and maintain resiliency.

*Trade-Mark

Refrigerator door pneumatically insulated with K-25 Fiber

Armstrong Cork Company

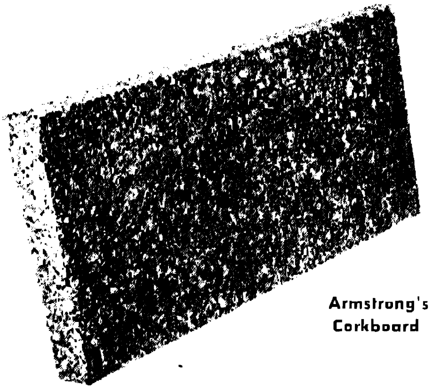
938 Concord Street, Lancaster, Pa.
Offices and distributors in principal cities

ARMSTRONG'S LOW-TEMPERATURE INSULATIONS

ARMSTRONG'S CORKBOARD

Thermal conductivity: 0.27 at 60° F.
Weight: Averages .60 to 0.65 lb. per bd. ft.
Sizes: 12" x 36", 18" x 36," 24" x 36", 36" x 36".
Thicknesses: 1", 1½", 2", 3", 4", and 6".

General: Armstrong's Corkboard is made entirely of pure cork granules, compressed into board form. This material is strong and tough yet easy to handle and install. It will not settle, sag, shrink, or warp; is fire-retardant, and will not harbor vermin. Portland cement plaster and asphalt emulsion finishes bond securely to corkboard surfaces. Armstrong's Corkboard conforms to Federal Specifications HH-C-561 b.

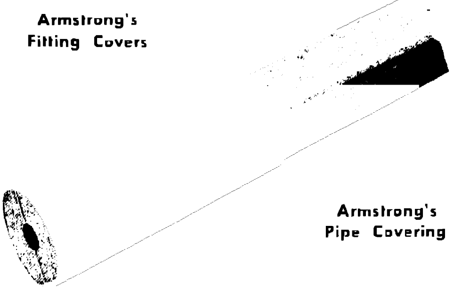


Armstrong's Corkboard

ARMSTRONG'S CORK COVERING

Armstrong's Cork Pipe Covering and Fitting Covers possess essentially the same physical characteristics as Armstrong's Corkboard. They are compressed and baked in molds, machined to accurate size, and finished with a heavy mastic coating which provides a seal against air and moisture penetration. Pipe covering is made in 36" half-sections, in Heavy, Standard, and Light-Duty thicknesses, for pipe sizes from ¼" up. Fitting covers are made in corresponding thicknesses for all types of ells, tees, valves, flanges, unions, etc. Armstrong's Cork Covering will also be made to customer's specifications for jobs requiring special sizes, thicknesses and shapes.

Armstrong's Fitting Covers



Armstrong's Pipe Covering

COMPLETE CONTRACT SERVICE

Armstrong's Contract Service provides materials, installation crews, and engineering assistance for all types of low-temperature insulation jobs. For complete information about the products and services offered by this nation-wide organization, contact your nearest Armstrong office or write direct to Armstrong Cork Company, 938 Concord Street, Lancaster, Pa.

American Flange & Manufacturing Co. Inc.

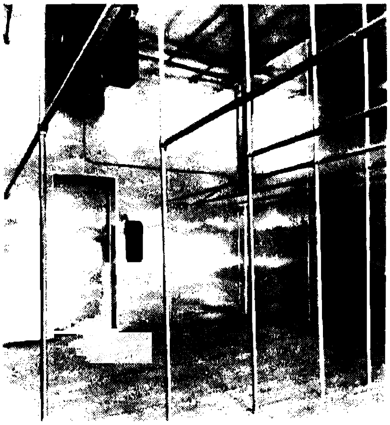
30 Rockefeller Plaza, New York 20, N.Y.
Plaza 7-2200

Ferro-Therm

Reg. U.S. Pat. Off.

STEEL INSULATION

Fully protected by U.S. and Foreign Patents
Issued and Pending



Ferro-Therm installed in a cold storage room

Ferro-Therm Steel Insulation, made from rigid steel sheets with a special alloy coating, reflects 90 per cent to 95 per cent of all radiant heat. This high reflectivity, combined with extremely low heat storage capacity, provides maximum insulating efficiency in minimum overall thickness.

Saves Pay Space and Weight

In cold storage construction, the number of sheets of Ferro-Therm depends on the temperature to be maintained and the U value required. The k value of Ferro-Therm, based on tests, is listed in the Data Book of the *American Society of Refrigerating Engineers* as 0.226 Btu per (hr) (sq ft) (°F temperature difference). Laboratory tests and thousands of applications have demonstrated that a wall of Ferro-Therm will provide insulating efficiency equivalent to a wall of mass insulation approximately twice as thick.

Assures Rapid Pull Down of Temperature

The low heat storage capacity of Ferro-Therm is extremely important in achieving rapid pull down of temperature, and in saving refrigeration costs for the initial and each subsequent cooling of space. Specifically, the heat storage capacity of a single sheet of No. 38 gauge is 0.029 Btu per (hr) (sq ft) (°F temperature difference). This is approximately 1/16 of the heat storage capacity of 1 sq ft of 1 in. thickness corkboard.

Permanent, Fire-Proof Insulation

Ferro-Therm construction eliminates trapping of moisture condensate, with subsequent deterioration of the construction. As it is all-metal, Ferro-Therm cannot be penetrated by rodents, vermin or termites, and is absolutely non-combustible. The value of Ferro-Therm for fire protection is apparent.

125° Below Zero Maintained in Altitude Test Chambers

Ferro-Therm has proved its superiority in buildings, cold storage rooms, refrigerated cabinets, locker rooms, dry ice containers, refrigerated railway car construction, ovens, high-temperature storage tanks—in fact, practically every type of application where high insulating efficiency with economies in space and weight are a requisite. The most notable demonstration of Ferro-Therm performance has been its selection for the insulation of altitude chambers for the testing of Army and Navy aviation equipment and personnel. In these chambers, temperatures as low as —125°F were maintained, with a temperature drop of +70°F to —70°F in 10 to 12 min.

INSULATION, CELLULOSE FIBRE (Continued)

Robt. A. Keasbey Co., 130 W. 19th St., N.Y.C. 11
Kimberly-Clark Corp., Neenah, Wis.
Masonite Corp., 111 W. Washington St., Chicago 2, Ill.
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Plastergon Wall Board Co., Box 40, Sta. B, Buffalo 7, N.Y.
Wood Conversion Co., First Nat'l. Bank Bldg., St. Paul 1, Mass *(p. 131)*

INSULATION, CORK

Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Cork Import Corp., 1 Engle St., Englewood, N.J.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
Delgro Mfg. Co., Inc., Boonton, N.J.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Albert O. Lloveras, 143 Waverly Place, N.Y.C. 14
Mundet Cork Corp., 7101 Tonnelle Ave., N. Bergen, N.J. *(p. 138)*
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 *(p. 136)*
United Cork Cos., Kearny, N.J. *(p. 139)*

INSULATION, COTTON

Gilman Bros. Co., Gilman, Ct.
Lockport Cotton Batting Co., Lockport, N.Y.

INSULATION, EXPANDED VERMICULITE

Ehret Magnesia Mfg. Co., Valley Forge, Pa
Grant Wilson, Inc., 315 S. Sherman Ave., Chicago 4, Ill.
Great Lakes Carbon Corp., 18 E. 48th St , New York 17, N.Y.
International Vermiculite Co., Girard, Ill.
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Western Mineral Products Co., 1720 Madison St., N.E., Minneapolis 13, Minn.
Zonolite Co., 135 S. La Salle St., Chicago 3, Ill.

INSULATION, FIBERGLAS

Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 128)*

INSULATION, FOAM GLASS

Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Pittsburgh Corning Corp., 307-4th Ave., Pittsburgh 22, Pa. *(p. 129)*
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Standard Asbestos Mfg. Co., 860 W. Evergreen Ave., Chicago 22, Ill.

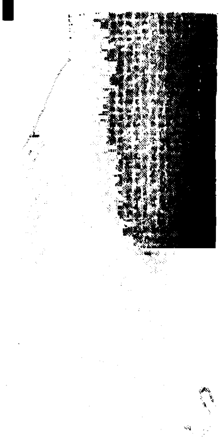
INSULATION, HAIR PRODUCTS

American Felt Co., Glenville, Ct.
American Hair & Felt Co., 222 N. Bank Dr., Chicago 54, Ill. *(p. 134)*
Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Grant Wilson, Inc., 315 S. Sherman Ave., Chicago 4, Ill.
Johns-Manville, 22 E 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Standard Asbestos Mfg. Co., 860 W. Evergreen Ave., Chicago 22, Ill.

INSULATION, MINERAL WOOL

Alton Mineral Wool Insulation Co., 2317 Whitemore Place, St. Louis 4, Mo.
Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Baldwin-Hill Co., 500 Breunig Ave., Trenton 2, N.J.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Carney Co., Inc., 153 Chestnut St., Mankato, Minn.
Cork Import Corp., 39 Park Place, Englewood, N.J.
Eagle-Picher Sales Co., American Bldg., Cin'. 1, O.

(Continued)



OZITE

INSULATING FELT

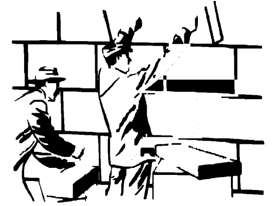
For over 50 years, all-hair OZITE insulating felt has set the pace for efficiency in cutting power costs . . . stopping heat losses, and providing low-temperature control.

You can get all the facts by requesting Bulletin No. 300—send for a copy today.

AMERICAN HAIR & FELT COMPANY
Dept. RDB1, Merchandise Mart, Chicago 54, Ill.

ROCK CORK

. . . No. 1 Cold Insulation for Food and Beverage Plants



- **SANITARY and ODORLESS**
- **MOISTURE-RESISTANT**
- **IMMUNE TO VERMIN**
- **PROVEN LONG LIFE**

Here's a top value, long lasting insulation for your cold rooms refrigerating equipment and piping It's Johns Manville Rock Cork* a basically mineral low temperature insulation composed of mineral wool combined in production with an asphaltic binder

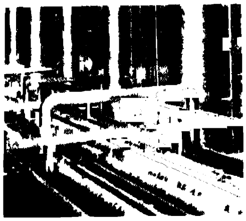
Because it is immune to termites will not support mold growth will not attract vermin or rodents and safeguards even the most sensitive food products from offensive odors Rock Cork is the No 1 choice of leading refrigeration engineers in the Food and beverage industries

Its firm fibrous nature insures thin, tight joints ... Rock Cork is easily handled and installed in any type of construction

ZEROLITE

. . . A New Inorganic Insulation For Many Refrigeration Services

- **HIGH FIRE RESISTANCE**
- **LOW HEAT CONDUCTIVITY**
- **MOISTURE-RESISTANT**
- **IMMUNITY TO MANY SOLVENTS**



In the oil and chemical industries Johns Manville Zerolite* is the outstanding cold insulation for tanks equipment and pipes

It is recommended where the fire problem is dominant where solvent fumes may cause costly deterioration of an insulation and where low heat conductivity is essential

Zerolite has the same valuable characteristics as Rock Cork PLUS the added advantage of 6 to 10 percent lower conductivity this means increased efficiency where large areas are to be treated

HOW ROCK CORK AND ZEROLITE ARE FURNISHED

Type	Length	Width	Thickness
Sheet	18"	18"	1"
	36"	18	1½ through 4"
Lagging**	18	2" through 6	1½" through 4
Pipe Insulation	3 ft section	to fit all standard pipe sizes	Ice Water Brine Heavy Brine

**Discs in the same thickness are available with max diam of 36"

For complete details on Rock Cork and Zerolite call your nearest Johns Manville office or write to Johns Manville, 22 East 40th Street New York 16 N Y

Rock Cork and Zerolite are registered Johns Manville Trade Marks

JOHNS MANVILLE

JM

PRODUCTS

Johns-Manville

22 EAST 40TH ST. NEW YORK 16, N. Y.

OFFICES IN ALL LARGE CITIES

INSULATION, MINERAL WOOL (Continued)

Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Forty-Eight Insulations, Inc., Aurora, Ill.
Grant Wilson, Inc., 315 S. Sherman Ave., Chicago 4, Ill.
Johns-Manville, 22. E. 40th St., N.Y.C. 16 (*p. 135*)
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Mundet Cork Corp., 7101 Tonnelle Ave., N. Bergen, N.J. (*p. 138*)
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y.
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg. Toledo 1, O. (*p. 128*)
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Standard Asbestos Mfg. Co., 860 W. Evergreen Ave., Chicago 22, Ill.

INSULATION, PIPE & TUBE (See also specific type of INSULATION)

Armstrong Cork Co., Lancaster, Pa. (*p. 132*)
Cork Import Corp., 1 Engle St., Englewood, N.J.
Engle-Picher Sales Co., American Bldg., Cin'ti 1, O.
Forty-Eight Insulations, Inc., Aurora, Ill.
Grant Wilson, Inc., 315 S. Sherman, Chicago 4, Ill.
International Vermiculite Co., Girard, Ill.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. (*p. 85*)
Johns-Manville, 22 E. 40th St., N.Y.C. 16. (*p. 135*)
Robt. A. Keasley Co., 139 W. 19th St., N.Y.C. 11
J. W. Mortell Co., Burch St., Kankakee, Ill.
Mundet Cork Corp., 7101 Tonnelle Ave., N. Bergen, N.J. (*p. 138*)
Mystik Adhesive Products, Div. of Chicago Show Printing Co., 2635 Kildare Ave., Chicago 39, Ill.
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y.
North Penn Co., 72-5th Ave., N.Y.C. 11
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. (*p. 128*)
Pittsburgh-Corning Corp., 307-4th Ave., Pittsburgh 22, Pa. (*p. 129*)
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 (*p. 136*)
Ric-Wil Co., Union Commerce Bldg., Cleveland 14, O.
Sponge Rubber Products Co., 1600 Derby Place Shelton, Ct.
Standard Asbestos Mfg. Co., 860 W. Evergreen Ave., Chicago 22, Ill.
Taft-Jenkins Co., Inc., 27 Sargeant St., Holyoke, Mass.
United Cork Cos., Kearny, N.J. (*p. 139*)

INSULATION, PLASTIC FOAM

Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.

INSULATION, REDWOOD BARK

Pacific Lumber Co., 100 Bush St., San Francisco 4, Cal. (*p. 137*)
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 (*p. 136*)

INSULATION, REFLECTIVE TYPE

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
American Flange & Mfg. Co., 30 Rockefeller Plaza, N.Y.C. 20 (*p. 133*)
Forty-Eight Insulations, Inc., Aurora, Ill.
C. T. Hogan & Co., Inc., 383 Madison Ave., N.Y.C. 17
Infra Insulation, Inc., 10 Murray St., N.Y.C. 7
Kimberley-Clark Corp., Neenah, Wis.
Permanente Products Co., 1924 B'way, Oakland 12, Cal.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
Sisalkraft Co., 205 W. Wacker Drive, Chicago 6, Ill.

INSULATION CLIPS

Lexington Supply Co., 4815 Lexington Ave., Cleveland 2, O.
Miracle Adhesives Corp., 214 E. 53rd St., N.Y.C. 22
Stic-Klip Mfg. Co., 50 Regent St., Cambridge 40, Mass. (*p. 127*)

INSULATION ERECTION MATERIALS

American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Armstrong Cork Co., Lancaster, Pa. (*p. 132*)
Cork Import Corp., 1 Engle St., Englewood, N.J.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
International Balsa Corp., 96 Boyd Ave., Jersey City 4, N.J.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Miracle Adhesives Corp., 214 E. 53rd St., N.Y.C. 22
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Rector Mineral Trading Corp., 164 E. 43rd St., N.Y.C 17 (*p. 136*)
Rubatex Div., Great American Industries, Bedford, Va. (*p. 130*)
Stancal Asphalt & Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Stic-Klip Mfg. Co., 50 Regent St., Cambridge 40, Mass. (*p. 127*)
United Cork Cos., Kearny, N.J. (*p. 139*)

INTERCOOLERS

Acme Industries, Inc., Mechanic & Ganson St., Jackson, Mich. (*p. 211*)
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago (*p. 49*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)

(Continued)

CORK BOARD

IS STILL THE BEST

INSULATION

THE BEST DOMESTIC AND IMPORTED
CORK FOR PROMPT DELIVERY

●

**STEAMBAKED—LIVES UP TO
FEDERAL SPECIFICATIONS**

●

SUBSTANTIAL PRICE SAVINGS DUE TO DIRECT SHIPMENTS FROM FOREIGN FACTORIES TO YOU.

*Excellent Opportunity For
Distributors & Manufacturers Agents*

Request details on open territories

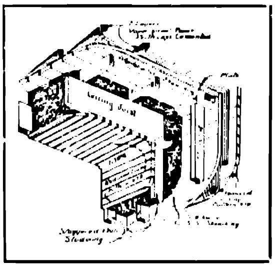
**RECTOR INSULATIONS
RECTOR MINERAL TRADING CORP.**
16 EAST 43rd STREET
NEW YORK 17, N.Y.
MUrray Hill 2-7912

Write, Wire or Phone Today!

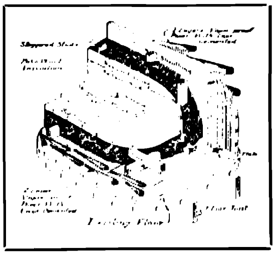
Specify PALCO INSULATION WOOL

FOR UNIFORM LOW TEMPERATURE CONTROL

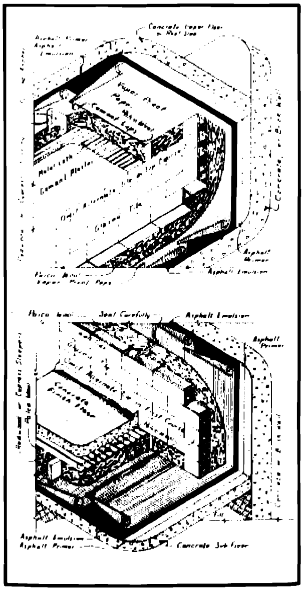
WHEN building or expanding frozen food and cold storage plants, experienced refrigeration engineers know PALCO WOOL Insulation meets the most rigid and exacting requirements of non-settling permanence, long-life endurance, low thermal conductivity and fire resistance. Produced from the bark of California Redwoods, PALCO WOOL Insulation is inherently moisture resistant, odor proof, and vermin repellent. Readily adaptable to all types of construction, PALCO WOOL Insulation not only keeps operating costs at a minimum through its high efficiency, but also effects savings in capital charges by reducing plant investment and depreciation. Investigate today how to get continuing economies with PALCO WOOL Insulation. Write for full details.



Typical Wall and Ceiling Wood Finish Construction for Low Temperatures.



Normal Frame Construction, Wood Walls, Flooring, and Exterior Siding for Cold Storage.

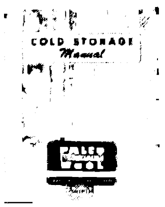


Construction detail for masonry floor, walls and ceiling—tile finish.

CHECK THESE SUPERIOR QUALITIES OF PALCO WOOL INSULATION

- √ Low Thermal Conductivity
- √ Durable
- √ Moisture Resistant
- √ Vermin Repellent
- √ Non-Settling
- √ Flame-Proofed
- √ Odor-Proof
- √ Economical

For Typical Construction Details Write Today for the New Cold Storage *Manual*



THE PACIFIC LUMBER COMPANY

100 BUSH STREET, SAN FRANCISCO 4, CALIFORNIA
35 E. WACKER DRIVE, CHICAGO 1, ILLINOIS

Jointite

TRADE MARK

COLD INSULATION

MUNDET CORKBOARD

MUNDET CORK PIPE COVERING

Mundet Products

"Jointite" Corkboard for flat cold insulation, "Jointite" Moulded Cork Pipe Covering and Fitting Covers for cold pipe lines, "Jointite" Cork Lagging, Mundet Natural Cork Isolation Mats for preventing transmission of vibration, "Jointite" Cork Roof insulation and all varieties of high grade cork specialties 85% Magnesia Pipe Covering and Block Insulation

Mundet "Jointite" Corkboard

Fabricated from 100% pure Cork—complies with U S Government Master Specifications Unsurpassed for all cold insulation services cold storage rooms, ice tanks, sharp freezers, roofs, refrigerators, refrigerator cars and trucks Recommended for acoustical correction Manufactured in only one grade "Jointite" is the most economical because it is the highest grade corkboard obtainable Standard size 12" x 36", thickness ½", ¾", 1", 2", 3", 4" and 6" Estimates furnished with or without our installation

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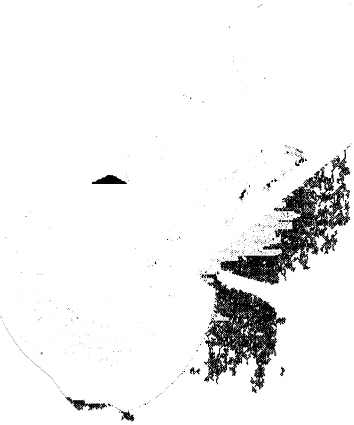
Quickly applied, the most economical cold pipe insulation on a value basis Mineral rubber finish gives protection from moisture or frost



View of Mundet "Jointite" Cork Pipe Covering, with moulded fitting cover

Made in 3 thicknesses, for all types and sizes of cold lines carrying sub zero to 50° temperature In sections 36" long with standard fitting covers

Mundet "Jointite" Beveled Cork Lagging

Cut to order to any desired radius, effectively insulates all types of cylindrical tanks and coolers

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7101 Tonnelle Ave. Insulation Division North Bergen, N J.

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UNITED is the pioneer for furnishing and making a complete installation of Cork insulation. For almost half a century our organization of specialized insulation engineers, trained in the school of practical heat, cold, sound and vibration insulation, have installed Corkboard in plants of every description, no matter how small or large.

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Manufactured in standard size boards 18" w. x 36" l., and in thicknesses of 1, 1½, 2, 3, 4 and 5 inches. Other thicknesses and widths—apply to factory.

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For brine, ammonia, ice or cold lines. Pipe covering and fitting jackets are molded to fit absolutely tight.

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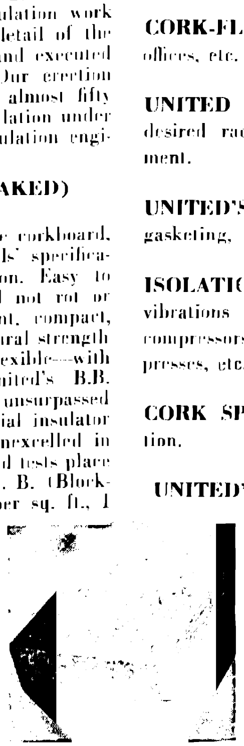

UNITED'S "Acousticork" is usually applied 1½ in. thick. Has high sound absorption co-efficiency. Combines noise quieting with high insulating value. Easy to install. Structurally strong. Can be painted to harmonize with decorative scheme. High light reflection when painted. Flexible.

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Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Trane Co., La Crosse, Wis. *(p. 118)*
York Corp., York, Pa. *(p. 52)*

IRON, ENAMELING (See SHEET, ENAMELING)

IRON PIPE COILS (See PIPE COILS, BENDS & BENDING)

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Henry Valve Co., Melrose Park, Ill. *(p. 141)*

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Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 204)*

LIQUID LEVEL CONTROLS (See CONTROLS, LIQUID LEVEL; also HIGHSIDE FLOATS; also LOWSIDE FLOATS; also FLOATS; also FLOAT SWITCHES)

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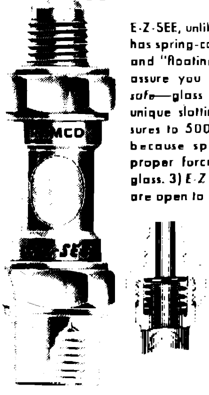
LIQUID INDICATORS
To Insure GREATER PROFITS



E-Z-SEE, unlike conventional liquid indicators, has spring-compensated, leak-proof gaskets and "floating" high pressure Pyrex glass to assure you these advantages: 1) *Perfectly safe*—glass protected from damage by unique slotting arrangement, safe at pressures to 500 Psi. 2) *Leak Proof*—can't leak because springs automatically maintain proper force to form positive seal around glass. 3) *E-Z to-see-thru*—both sides of body are open to let in light through tubular glass.

Available in flare sizes ¼″ thru ⅝″ with both ends male, one end male and one end female and in OD sweat sizes ⅜″ thru 1⅝″.

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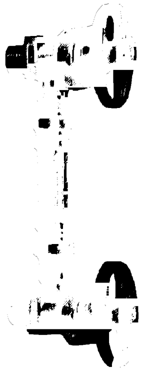

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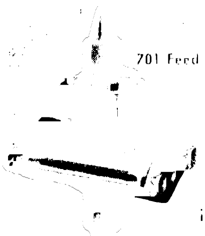
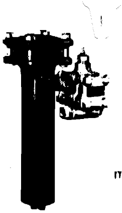
FLOAT CONTROLS



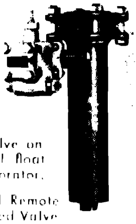
701 Feed Valve
Capacities Up to 2000 Tons.

The No. 701 feed valve operates by means of the varying pressure in a power cylinder above the piston. Perfect modulated action obtained on flooded systems when used in conjunction with either No. 300 HP or No. 100 VP float controls. No. 701 Feed Valve may also be used with thermal expansion or magnetic solenoid.



100 VP Pilot Float Valve
Welded Steel Chamber

300 HP Pilot Float Valve



The No. 101 Remote Feed Valve on the Right permits location of float and chamber outside of evaporator. Adjustable liquid level.

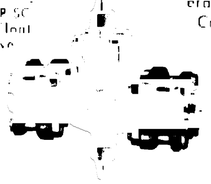
101 Remote Feed Valve With Float Chamber

FOR "FREON"

The No. 701 Feed Valve is a pilot controlled piston valve actuated by pressures in the power cylinder. Because of its quick and responsive action, the 205 VP-SC Pilot is recommended for ideal modulated action of refrigerant.

Capacities Up to 1000 Tons.

250 VP SC Pilot Float Valve



701 "Freon" Feed Valve

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DESIGNERS AND ENGINEERS
REFRIGERATION CONTROL SYSTEMS

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Magnetrol, Inc., 2110 S. Marshall Blvd., Chicago 23, Ill.
(A,B) H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. (*p. 142*)
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
(A) Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
(A,B) York Corp., York, Pa. (*p. 52*)

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Frick Co., Waynesboro, Pa. (*p. 51*)
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METERS, ELECTRIC

Bristol Co., Waterbury 20, Ct. (Recording)
R. W. Cramer Co., Inc., Centerbrook, Ct.
Esterline-Angus Co., Inc., P.O. Box 596, Indpls. 6, Ind.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
J-B-T Instruments, Inc., 441 Chapel St., New Haven 8, Ct.
Sangamo Elec. Co., Springfield, Ill.
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

METERS, FLOW

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Badger Meter Mfg. Co., Milwaukee 10, Wis.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., P.O. Box 1342, Providence 1, R.I.
Fischer & Porter Co., Hatboro, Pa.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Claud S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
Hastings Instrument Co., Inc., Super Highway & Pine Ave., Hampton, Va.
Hays Corp., Michigan City, Ind.
Henszey Co., 202½ N. Water St., Watertown, Wis.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Neptune Meter Co., 50 W. 50th St., N.Y.C. 20
Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Schutte & Koerting Co., Cornwells Heights, Bucks County, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Worthington Pump & Machinery Corp, Harrison, N.J. *(p. 144)*

METERS, RECORDING

Badger Meter Mfg. Co., Milwaukee 10, Wis.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., P.O. Box 1342, Providence 1, R.I.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Esterline Angus Co., Inc., P.O. Box 596, Indpls. 6, Ind.
Fischer & Porter Co., Hatboro, Pa. (Flow Rate)
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Hays Corp., Michigan City, Ind.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Tagliabue Instruments Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.

METERS, RUNNING TIME

Bristol Co., Waterbury 20, Ct.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
R. W. Cramer Co., Box 25, Centerbrook, Ct.
J-B-T Instruments, Inc., 441 Chapel St., New Haven 8, Ct.
Westinghouse Electric Corp., Plane & Orange Sts., Newark 1, N.J.

Worthington Pump and Machinery Corp.

Air Conditioning and Refrigeration Division

General Offices: Harrison, N.J.

District Offices and Representatives in Principal Cities

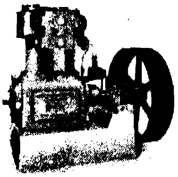
WORTHINGTON

A-1-33

Refrigeration Systems for Air Conditioning Industrial Process and All Other Applications

Complete refrigerating systems for use with Freon-11, Freon-12, Methyl Chloride, Ammonia or Carbon Dioxide, either direct expansion or water cooling applications. A complete line of refrigeration compressors, permitting impartial recommendations. A nation-wide organization of distributors in major cities to provide sales and engineering service and to plan complete air conditioning systems of the central or unit type. Architects, engineers and contractors are invited to consult with us. Write to Harrison, N.J., or to anyone of our district offices for information on these products.

VERTICAL AMMONIA COMPRESSORS



Pressure-lubricated, roller main bearings; safety heads; patented Feather Valves; belt drive or direct connected to electric motor, diesel or gas engine; sizes 3 x 3 to 10 x 10, 2-cylinder.

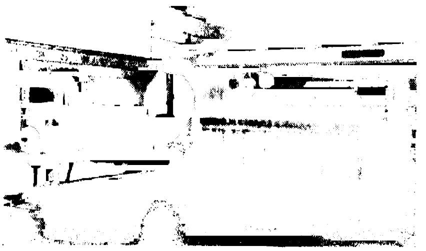
SELF-CONTAINED FREON-12 CONDENSING UNITS

Capacities from 3 to 125 tons. Condensers are highly efficient multi-pass shell and straight tube type. Condenser heads readily removable, making tubes easily accessible for cleaning. Small sizes have condensers mounted in base; larger units have condenser mounted above compressor and motor. Minimum space required.

HORIZONTAL AMMONIA COMPRESSORS

Single and duplex; single-stage and two-stage; belt drive or direct connected to electric motor, diesel or steam engine; patented Feather valves; ratings from 60 to 1000. Automatic capacity control features are easily applied.

ICE PLANTS



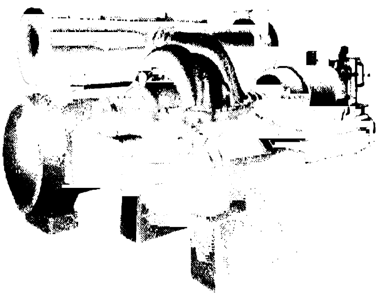
Ice plant equipment of all types, including complete refrigeration machines with Worthington gas engine, diesel engine or steam drive in one unit. Also, motor or belt driven compressors. Ice tank with loose can or group lift. High transfer coils in brine race or brine coolers as best suited.

Ammonia Absorption Refrigeration Machinery

All Types of Auxiliary Equipment, Condensers, Coolers, Fittings, Etc.

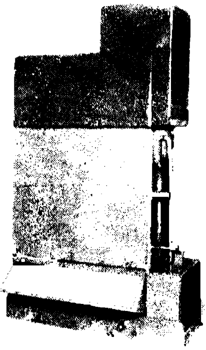
Worthington Pump and Machinery Corp.

CENTRIFUGAL REFRIGERATION WATER COOLING SYSTEMS



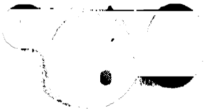
Freon centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive. 56 unit sizes . . . 150 to 1200 tons.

EVAPORATIVE CONDENSERS



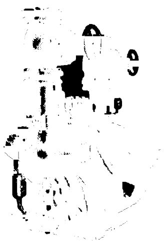
Series ECZ Freon-12 illustrated. 10 to 150 tons refrigeration. Units of sectionalized construction; all parts easily accessible. Galvanized steel coils for ammonia. Bare copper coils for Freon-12.

CONDENSERS AND BRINE COOLERS



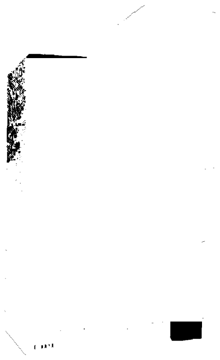
Multi-pass, as illustrated for closed systems and space saving. Heads efficiently baffled to produce high liquid velocity. Vertical "Spira-Flo" types provided with circular water box and special water distributors. Highly efficient, easily accessible for cleaning without interfering with condenser operation.

BOOSTER REFRIGERATION COMPRESSORS



Multi-cylinder single-acting for capacities up to 400 cfm. Horizontal double-acting for larger capacity requirements. Designed to produce and maintain low evaporator pressures required for low temperature applications in conjunction with compression or absorption refrigeration systems.

PRODUCT COOLERS



Dry and brine spray type units in many sizes and arrangements. Capacities from 2 to 37 tons. Air circulation rates from 2000 to 16,000 cfm. Sectionalized construction with removable panels. Finned or bare pipe coils for use with Freon or Ammonia.

AMMONIA VALVES AND FITTINGS

Smaller sizes of all steel construction. Larger size valve bodies, ells and tees, close grained iron.

METHYL CHLORIDE & METHYLENE CHLORIDE

Ansul Chemical Co., Marinette, Wis. *(p. 115)*
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal.
Virginia Smelting Co., W. Norfolk, Va. *(p. 30)*

MILK COOLERS, CABINET TYPE

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Esco Cabinet Co., West Chester, Pa.
Evans Mfg. Corp., 460 S. 10th St., Mt. Vernon, N.Y.
Frigidaire Div., Gen'l Motors Corp., Dayton 1, O. *(p. 17)*
Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6, Mo.
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Emil Steinhorst & Sons, Inc., 612 South St., Utica N.Y.
United Refrigerator Co., Hudson, Wis.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Westinghouse Elec. Corp., 653 Page Blvd., Springfield 2, Mass.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

MILK COOLERS, SURFACE TYPE (See WATER COOLERS, BAUDELOT TYPE)

MIRRORS

Hooker Glass & Paint & Mfg. Co., 659 W. Washington Blvd., Chicago, Ill.
Semon Bache & Co., 636 Greenwich St., N.Y.C. 14

MOISTURE INDICATORS

Davison Chemical Corp., Baltimore 3, Md. *(p. 81)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*

MOLDED PRODUCTS, PLASTIC

Anchor Plastics Co., Inc., 533 Canal St., N.Y.C. 13
Bakelite Corp., 30 E. 42nd St., N.Y.C. 17
Cambridge Molded Plastics Co., Cambridge, O.
Chicago Molded Products Corp., 1020 N. Kolmar Ave., Chicago, Ill.
Crescent Plastics, 303 1st Ave., Evansville, Ind.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis
General American Transportation Corp., 135 S. La Salle St., Chicago 90, Ill.
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
General Electric Co., 100 Woodlawn Ave., Pittsfield Mass.
General Industries Co., Olive & Taylor St., Elyria, O
Gits Molding Corp., 4600 W. Huron St., Chicago 1, Ill
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind.
Kent Plastics Corp., 1528 N. Fulton Ave., Evansville, Ind.
Mack Molding Co., Arlington, Vt.
National Lock Co., Rockford, Ill.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Plaskon Div., L-O-F Glass Co., Sylvan Ave., Toledo, O
Prolon Plastics, Florence, Mass.
Standard Molding Corp., 1517 E. 3rd St., Dayton 1, O.
Amos Thompson Corp., Edinburg, Ind.
Wolverine Plastics, Inc., 495 Redman Rd., Milan, Mich

MONEL (See also mill forms, i.e., SHEET, etc.)

International Nickel Co., 67 Wall St., N.Y.C. 5

MONORAILS (See CONVEYORS)

MORTUARY REFRIGERATORS (See REFRIGERATORS, MORTUARY)

MOTOR-GENERATORS

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Crocker-Wheeler Div., Joshua Hendy Iron Wks., Ampere, N.J.
Elliott Co., Jeannette, Pa.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hobart Bros. Co., 146 Hobart Square, Troy, O.
Ideal Electric, E. First St., Mansfield, O
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Marathon Elec. Mfg. Corp., Wausau, Wis
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Robbins & Myers Inc., 1345 Lagonda Ave., Springfield, O.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Westinghouse Elec. Corp., 4451 Genesee St., Buffalo 5 N.Y.

MOTOR BASES, BRACKETS & SUPPORTS (See BASES)

MOTOR PROTECTORS

Chase-Shawmut Co., Newburyport, Mass
Spencer Thermostat Div., Metals & Controls Corp., 34 Forest St., Attleboro, Mass

MOTOR STARTERS (See STARTERS)

MOTORS

Alliance Mfg. Co., 100 Lake Park Blvd., Alliance, O
Louis Allis Co., 127 E. Stewart St., Milwaukee 7, Wis
Allis-Chalmers Mfg., Co., Milwaukee 1, Wis.
Baldor Elec. Co., 4353 Duncan Ave., St. Louis 10, Mo
Barber Coleman Co., Rockford, Ill
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4411 Wayne Ave., Phila. 44, Pa
Burke Elec. Co., Erie, Pa
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O
Cleveland Elec. Motor Co., 5213 Chester Ave., Cleveland 3, O.
Continental Elec. Co., Inc., 325 Ferry St., Newark 5, N.J.
R. W. Cramer Co., P.O. Box 25, Centerbrook, Ct.
Crocker-Wheeler Div., Joshua Hendy Iron Wks., Ampere, N.J
Delco Products Div., Gen'l. Motors Corp., 329 E. 1st St Dayton, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Doerr Electric Corp., Cedarburg, Wis
Eastern Air Devices, Inc., 130 Flatbush Ave., Brooklyn, N.Y.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro N.J.
Electric Boat Co., Ave. A & North St., Bayonne, N.J
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
Electric Motor Corp., 1211 State St., Racine, Wis.
Elliott Co., Jeannette, Pa.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Fasco Industries, Inc., Union & Augusta Sts., Rochester 2, N.Y.
Forbes & Myers, 173 Union St., Worcester, Mass.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Haydon Mfg. Co., 245 E. Elm St., Torrington, Ct.
Hobart Bros. Co., 146 Hobart Square, Troy, O.
Hoover Elec. Co., 2100 Stoner Ave., W. Los Angeles 25, Cal
Howell Elec. Motors Co., 409 N. Roosevelt, Howell, Mich.
Ideal Elec. & Mfg. Co., Mansfield, O.

Imperial Elec. Co., 84 Ira St., Akron, O.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Kimble Elec., Div. of Miehle Printing Press & Mfg. Co., 2850 Mt. Pleasant St., Burlington, Ia.
King-Wyse, Inc., Archbold, O.
Kingston-Conley Elec. Co., 86 Brook Ave., N. Plainfield, N.J.
Leland Elec. Co., 1501 Webster St., Dayton 1, O.
Lima Elec. Motor Co., Findlay Rd., Lima, O.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Marble-Card Elec. Co., Gladstone, Mich.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Moore Co., 544 Westport Rd., Kansas City 2, Mo.
Ohio Gear Co., 1333 E. 179 St., Cleveland, O.
Onsrud Machine Wks., Inc., 3900 Palmer St., Chicago 47, Ill.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Philadelphia 20, Pa
Piqua Machine & Mfg. Co., Young & Coolidge Sts., Piqua O.
Redmond Co., Inc., Owosso, Mich.
Reliance Elec. & Engrg. Co., 1088 Ivanhoe Rd., Cleveland 10, O.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Robbins & Myers, Inc., Springfield, O.
Signal Elec. Mfg. Co., Menominee, Mich.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1, Wis.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Sterling Elec. Motors, Inc., 5401 Anaheim-Telegraph Rd., Los Angeles 22, Cal.
O. A. Sutton Corp., 1812 W. 2nd St., Wichita 1, Kan.
U. S. Elec'l. Motors, Inc., Los Angeles 54, Cal. & Milford, Ct.
Valley Electric Corp., 4221 Forest Park Blvd., St. Louis 8, Mo.
Victor Elec. Products, Inc., 2950 Robertson Rd., Cincinnati 9, O.
Wagner Elec. Corp., 6400 Plymouth Ave., St. Louis 14, Mo.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3. N.J.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Westinghouse Elec. Corp., Lima, O.

MOTORS, DAMPER (See DAMPER MOTORS)

MOULDINGS

Brasco Mfg. Co., Harvey, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.
Greene Mfg. Co., Inc., 1028 Douglas Ave., Racine, Wis.
John Lees Div., Serrick Corp., 1300 Batavia, Muncie, Ind.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber & Plastic)
R. D. Werner Co., Inc. 295-5th Ave., N.Y.C. 16

MOUNTINGS, VIBRATION ABSORBING (See VIBRATION ABSORBING BASES)

MUFFLERS, PIPE LINE

Alexander-Tagg Industries, Inc., Jacksonville Rd. & Summit Ave., Hatboro, Pa.
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Drico Industrial Corp., 29 Broadway, N.Y.C. 6
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*

NAILS

All-metal Screw Products Co., Inc., 33 Greene St., N.Y.C. 13
Aluminum Co. of America, Pittsburgh 19, Pa.
American Rolling Mill Co., Middletown, O. (Stainless Steel)
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Armco Steel Corp., Middletown, O.
Atlantic Steel Corp., P.O. Box 1714, Atlanta, Ga.
Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburgh 19, Pa. (Copper)
International Nickel Co., 67 Wall St., N.Y.C. 5 (Nickel Alloy)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky. (Aluminum)
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Townsend Co., 205 River Rd., New Brighton, Pa.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18
Alan Wood Steel Co., Steel Mill Rd., Conshohocken, Pa (Cut)
Youngstown Sheet & Tube Co., Youngstown, O.

NAME PLATES (See also TRIM)

Acme Metal Etching Co., Inc., 4857 St. Aubin, Detroit 7, Mich
Aeromark Co., 5 Morrell St., Elizabeth 5, N.J.
American Emblem Co., Inc., P.O. Box 116-H, Utica 1, N.Y.
Anderson & Sons, Inc., N. Elm St., Westfield, Mass.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Chandler Co., 1407 Park St., Hartford 6, Ct.
Chicago Thrift Etching Co., 1555 N. Sheffield Ave., Chicago, Ill.
Corning Glass Co., Corning N.Y.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Electro-Chemical Engraving Co., Inc., 1100 Brook Ave., N.Y.C.
Fox Co., Fox Lane, Cin'ti. 23, O.
General Industries Co., Olive & Taylor Sts., Elyria, O.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 160 Scribner Ave., N.W., Grand Rapids 1, Mich.
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
James H. Mathews & Co., 3948 Forbes St., Pittsburgh 13, Pa.
Geo. J. Mayer Co., 15 N. Pennsylvania St., Indpls. 9, Ind.
A.B. Murray, Inc., 604 Green Lane, Elizabeth, N.J.
Newman Bros., Inc., 666 W. 4th St., Cin'ti. 3, O.
Northern Engraving & Mfg. Co., 4th & Vine Sts., La Crosse, Wis.
Premium Plastics, Inc., 627 Salem Ave., Dayton 6, O.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Sneath Glass Co., Hartford City, Ind.
Standard Molding Corp., 1517 E. 3rd St., Dayton 1, O.
Standard Thermometer, Inc., 952 Dorchester Ave., Boston 25, Mass.
Amos Thompson Corp., Edinburg, Ind.

NEEDLE VALVE ASSEMBLIES

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
Conoflow Corp., 2100 Arch St., Phila. 3, Pa.
Fairbanks Co., 393 Lafayette St., N.Y.C. 3
Fedders-Quigan Corp., 57 Tonawanda St., Buffalo, N.Y.
Jerguson Gage & Valve, 87 Fellsway, Somervile, Mass.
Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.

NEEDLES, VALVE

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Illl.
Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.

NEGATIVE TEMPERATURE COEFFICIENT MATERIAL

Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.

NETTING, WIRE (See MESH, METAL)

NICKEL (See also mill forms, i.e., ROD; also SHEET, etc.)

Haynes Stellite Co., Unit. of Union Carbide & Carbon Corp., Kokomo, Ind.

International Nickel Co., 67 Wall St., N.Y.C. 5

NIPPLES, PIPE (See FITTINGS)

NITROGEN

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17

Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17

NOSES, SEAL

Chicago Seal Co., 332 S. Hoyne Ave., Chicago 20, Ill.

Clifford Mfg. Co., Grove St., Waltham, Mass.

Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.

NOZZLES

April Showers Co., 4126-8th St., N.W., Washington 11, D.C. (*p. 169*)

Bahnson Co., 1000 S. Marshall St., Winston-Salem, N.C.

Baliofet Dies & Nozzle Co., Inc., 6825 Adams St., Guttenberg, N.J.

Binks Mfg. Co., 3124-32 Carroll Ave., Chicago 16, Ill. (*p. 73*)

Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.

Clarage Fan Co., Porter St., Kalamazoo 16, Mich.

Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.

Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.

Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.

Grinnell Co., 260 W. Exchange St., Providence 1, R.I.

Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.

Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.

Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. (*p. 75*)

Jos. A. Martorello & Co., 229 N. 13th St., Phila. 7, Pa (*p. 147*)

Monarch Mfg. Wks., Inc., Salmon & Westmorland Sts., Phila. 34, Pa.

D. J. Murray Mfg. Co., 1024-3rd St., Wausau, Wis.

Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.

Parks-Cramer Co., Box 444, Fitchburg, Mass.

Phillips Cooling Tower Co., Inc., 220 DuPont St., Brooklyn 22, N.Y.

J. F. Pritchard & Co., 908 Grand Ave., Kansas City 6, Mo. (*p. 72*)

Rhode Island Humidifier & Ventilating Co., 99 Chauncey St., Boston 11, Mass.

Schutte & Koerting Co., Cornwells Heights, Bucks County, Pa.

Scoville Mfg. Co., 61 Mill St., Waterbury 20, Ct.

Spray Engrg. Co., 114 Central St., Somerville 45, Mass.

Spraying Systems Co., 3201 Randolph St., Bellwood, Chicago, Ill.

St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.

Supreme Elec. Products Co., 194 Vassar St., Rochester 7, N.Y.

Taylor Forge & Pipe Wks., P.O. Box 485, Chicago 90, Ill.

Thermal Industries, P.O. Box 725, Indio, Cal.

Water Cooling Corp., 71 Nassau St., N.Y.C. 7

Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo.

Wooster Brass Co., 1415 E. Bowman St., Wooster, O.

Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

NUTS (See BOLTS)

ODOR ELIMINATION UNITS & SYSTEMS (See AIR PURIFICATION EQUIPMENT)

OIL COOLERS & OIL COOLING SYSTEMS (See also COOLANT COOLERS)

Acme Industries, Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)

Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. (*p. 19*)

American District Steam Co., Bryant St., N. Tonawanda, N.Y.

Richard M. Armstrong Co., Box 188, W. Chester, Pa.

Black, Sivalls & Bryson, Kansas City 3, Mo.

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)

Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.

Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p. 31*)

Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa. (*p. 57*)

Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)

Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)

Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)

Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)

Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)

Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.

Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.

Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.

Schutte & Koerting Co., Cornwells Heights, Bucks County Pa.

Taco Heaters, Inc., 137 South St., Providence 3, R.I.

Trane Co., La Crosse, Wis. (*p. 118*)

Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.

P. S. Williams Co., Inc., 6 North St., Mount Vernon 1, N.Y.

Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

York Corp., York, Pa. (*p. 52*)

Young Radiator Co., Racine, Wis. (*p. 77*)

OIL ENGINES (See ENGINES, DIESEL)

OIL SEPARATORS

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. (*p. 149*)

Richard M. Armstrong Co., Box 188, W. Chester, Pa.

Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)

Black Sivalls & Bryson, Kansas City 3, Mo.

Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)

Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.

Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)

King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. (*p. 4*)

Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)

H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. (*p. 142*)

Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Philadelphia 46, Pa.

Rex Engineering & Sales Co., Box 5141, Oklahoma City, Okla.

Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.

Temprite Products Corp., E. Maple Rd., Birmingham, Mich. (*p. 89*)

Wabash Mfg. Co., 2300 S. Western Ave., Chicago 8, Ill.

Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

York Corp., York, Pa. (*p. 52*)

OILERS & OILING SYSTEMS (See LUBRICATORS)

OILS, LUBRICATING

Ansul Chemical Co., Marinette, Wis. *(p. 115)*
Atlantic Refining Co., 260 S. Broad St., Phila., Pa.
Carbide & Carbon Chemicals Co., Div. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Cities Service Oil Co. (Pa.), 60 Wall Tower, N.Y.C. 5
Esso Standard Oil Co., 15 W. 51st St., N.Y.C. 19
Fiske Bros. Refining Co., 129 Lockwood St., Newark 5, N.J.
Gulf Oil Corp., Gulf Bldg., Pittsburgh, Pa.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Keystone Lubricating Co., 21st & Lippincott Sts., Phila. 32, Pa.
Macmillan Petroleum Corp., 530 W. 6th St., Los Angeles 14, Cal.
New York & New Jersey Lubricant Co., 292 Madison Ave., N.Y.C. 17
Shell Oil Co., Inc., 50 W. 50th St., N.Y.C. 20
Sinclair Refining Co., 630-5th Ave., N.Y.C. 20
Socony-Vacuum Oil Co., 26 B'way, N.Y.C. 4
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y C. 16
Standard Oil Co. of California, 225 Bush St., San Francisco 20, Cal.
D. A. Stuart Oil Co., Ltd., 2727 S. Troy St., Chicago 23, Ill.
Sun Oil Co., 1608 Walnut St., Phila. 3, Pa.
Swan-Finch Oil Corp., 30 Rockefeller Plaza, N.Y.C. 3
Texas Co., 135 E. 42nd St., N.Y.C. 17
Tide Water Associated Oil Co., 17 Battery Place, N.Y.C. 4
Valvoline Oil Co., 534 E. 5th St., Cin'ti 2, O.

OZONE EQUIPMENT

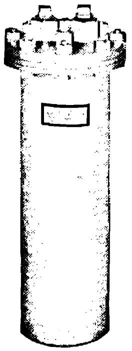
Electroaire Corp., 41-38-37th St., Long Island City 1, N.Y.
Norwood Filtration Co , Northampton, Mass.
Welsbach Corp., 1500 Walnut St., Phila. 2, Pa.

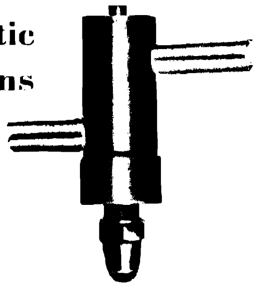
PACKING, SHAFT, VALVE, etc. (See GASKETS & GASKET MATERIAL)

PAINT, ASPHALT, CORROSION RESISTING, HEAT RESISTING, etc. (See also FINISHES)

Aluminum Industries, Inc., 2438 Beekman St., Cincinnati 25, O.
Amercoat Div., American Pipe & Construction Co., 4809 Firestone Blvd., Southgate, Cal.
American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
American Chemical Paint Co., Ambler, Pa.
American Pipe & Construction Co., Box 3428 Terminal Annex P.O., Los Angeles, Cal.
Arco Co., 7301 Bessemer Ave., Cleveland 27, O.
Atlas Mineral Products Co., Mertztown, Pa.
Atlas Powder Co., North Chicago, Ill.
Baer Bros., 438 W. 37th St., N.Y.C. 18
Calbar Paint & Varnish Co., 2612 N. Martha St., Phila. 25, Pa.
Dearborn Chemical Co., Merchandise Mart Plaza, Chicago 54, Ill.
Dennis Chemical Co., 2701 Papin St., St. Louis 3, Mo.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Ernecke & Salmstein Co., 1611 N. Sheffield Ave., Chicago 14, Ill.
Esso Standard Oil Co., 15 W. 51st St., N.Y.C. 19
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Glidden Co., 11001 Madison Ave., Cleveland 2, O.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Alfred Hague & Co., 227-34th St., Brooklyn 32, N.Y.
Hooker Glass & Paint Mfg. Co., 659 W. Washington Blvd., Chicago, Ill.
A. C. Horn Co., Inc., 10th St. & 44th Ave., Long Island City 1, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Interchemical Corp., 57 State St., Newark, N.J.
Jones-Dabney Co., Div. of Devoe & Raynolds, 1481 S. 11th St., Louisville 8, Ky.

(Continued)





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Oil Separators with Automatic Oil Return 1/3 H.P. to 120 Tons

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DETROIT 3, Michigan

PAINT (Continued)

LKR Chemical Products Co , 3113 Park Ave , Detroit 1, Mich
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Master Mechanics Co., Freeman Rd., Cleveland 13, O.
Midland Paint & Varnish Co., 3801 E. 91st St., Cleveland 5, O.
National Lead Co., 111 Broadway, N.Y.C. 6
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20. O.
Nukem Products Corp , 111 Colgate Ave , Buffalo 20, N.Y.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Randolph Products Co., Carlstadt, N.J. *(p 103)*
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Rutland Fire Clay Co., Curtis Ave., Rutland, Vt.
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Stancal Asphalt & Bitumuls Co., 200 Bush St., San Francisco 22, Cal.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
United Chromium, Inc , 100 E. 42nd St., N.Y.C. 17
U. S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Wailes Dove-Hermiston Corp., 448 South Ave., Westfield, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
J C Whitlam Mfg. Co , Wadsworth, O.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Zapon Div., Atlas Powder Co., Stamford, Ct.

PANELS, INSTRUMENT (See INSTRUMENT PANELS)

PAPER, ASBESTOS

Alfol Div , Reflectal Corp , 155 E 44th St N Y C 17
A. W. Chesterton Co , 4 Ashland St., Everett 49, Mass
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C 11
Ruberoid Co , 500-5th Ave , N.Y.C 18
Standard Asbestos Mfg Co , 860 W Evergreen Ave , Chicago 22, Ill
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
Grant Wilson, Inc , 315 S Sherman, Chicago 4, Ill.

PAPER, BUILDING

Aetna Plywood & Veneer Co., 1799 Elston Ave., Chicago 21, Ill.
Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Angier Corp., Framingham, Mass.
Atlas Asbestos Co., Ltd., 110 McGill St , Montreal 1, Quebec, Canada
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Presstite Engrg. Co., 3900 Chouteau Ave , St. Louis 10, Mo.
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C 17 *(p. 136)*
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Grant Wilson, Inc., 316 S. La Salle St , Chicago 4, Ill.

PAPER & FABRIC, COATED OR IMPREGNATED (See also FIBRE)

E. I du Pont de Nemours & Co , Inc , Wilmington 98, Del
Formica Co., 4613 Spring Grove Ave., Cin'ti 32, O.
Panelyte Div , St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Wilmington Fibre Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.
Wolverine Fabricating & Mfg. Co., Inc., Princess St. & M.C.R.R., Inkster, Mich.

PAPER PRODUCTS, MOLDED

Panelyte Div., St. Regis Paper Co., 230 Park Ave , N.Y.C. 17

PASSING DOORS (See COLD STORAGE DOORS)

PERFORATED METALS (See METAL, PERFORATED)

PHOSPHOR BRONZE

American Brass Co., Waterbury 88, Ct. *(p 189)*
Bristol Brass Corp., Bristol, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury, Ct.
C G Hussey & Co , Inc., 2860 2nd Ave., Pittsburgh 19, Pa
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. *(p. 191)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Western Brass Mills, Div. of Olin Industries, Inc., E. Alton 17, Ill.

PHOTOGRAPHIC COOLING EQUIPMENT (See also WATER COOLERS)

Dole Refrigerating Co , 5910 N Pulaski Rd Chicago 30 Ill
Filtrine Mfg. Co., Inc., 53 Lexington Ave., Brooklyn 5, N.Y. *(p 90)*
Gay Engrg. Co , 2730 E 11th St Los Angeles 23, Cal
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p 120)*
Star Metal Mfg. Co Inc , Trenton Ave & Ann St , Philadelphia 34, Pa
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p 89)*
Westinghouse Elec Corp., Bloomfield N J
York Corp., York, Pa. *(p. [illegible])*

PILLOW BLOCKS (See BEARINGS)

PILOT LIGHTS

Drake Mfg. Co , 1709 W. Hubbard St , Chicago 22, Ill.
General Electric Co , 1 River Rd., Schenectady 5, N Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Amos Thompson Corp., Edinburgh, Ind.

PINS, DOWEL, etc.

Acme Industrial Co , 205 N. Laflin St., Chicago 7 Ill
Allmetal Screw Products Co., Inc , 33 Greene St., N.Y.C. 13
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Holo-Krome Screw Corp , 25 Brook St., Hartford 10, Ct.
Standard Pressed Steel Co., Jenkintown, Pa.
Standard Steel Specialty Co., Beaver Falls, Pa.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Townsend Co., New Brighton, Pa.

PIPE (See particular type following; also TUBES & TUBING)

PIPE, COPPER, BRASS, BRONZE, etc.

American Brass Co., Waterbury 88, Ct. *(p. 189)*
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburgh 19, Pa.
Lewin-Mathes Co., 1111 Chouetau. St. Louis 2, Mo.
Mueller Brass Co., Port Huron, Mich.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

PIPE, CORROSION RESISTING

Allmetal Screw Products Co., 33 Greene St., N.Y.C. 13
American Brass Co., Waterbury 88, Ct. (*p. 189*)
Atlas Mineral Products Co., Mertztown, Pa.
Babcock & Wilcox Tube Co., Beaver Falls, Pa.
Bethlehem Steel Co., Bethlehem, Pa.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Duriron Co., Inc., Dayton 1, O.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N J.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
National Lead Co., 111 B'way, N.Y.C. 6
National Tube Co., U. S. Steel Corp., Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

PIPE, FURNACE & STOVE

Excelsior Steel Furnace Co., 118 S. Clinton St., Chicago 6, Ill.
Henry Furnace Co., Medina, O.
I. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee 7, Wis.
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y.
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Reeves Steel & Mfg. Co., 137 E. Iron Ave., Dover, O.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Wheeling Corrugating Co., Wheeling, W. Va.
Wheeling Steel Corp., Wheeling, W. Va.

PIPE, HARD RUBBER & PLASTIC

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Haveg Corp., Marshallton, Del.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Luzerne Rubber Co., Trenton 9, N.J.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

PIPE, STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
W. R. Ames Co., 150 Hooper St., San Francisco 7, Cal.
Armco Drainage & Metal Products, Inc., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Central Iron & Steel Co., Harrisburg, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Laclede Stee Co., Arcade Bldg., St. Louis 1, Mo
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks. Inc. 128 Linden St., Allentown, Pa.
National Tube Co., U. S. Steel Corp., Subsidiary, P.O Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa. (Steel & Stainless)
Reeves Steel & Mfg. Co., 137 E. Iron Ave., Dover, O.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Taylor Forge & Pipe Wks., P.O. Box 485, Chicago 90, Ill.
Wheeling Steel Corp., Wheeling, W. Va.

PIPE, WROUGHT IRON

A. M. Byers Co., Box 1076, Pittsburgh 30, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.

PIPE CLAMPS & HANGERS (See CLAMPS & HANGERS)

PIPE COILS, BENDS & BENDING (See also RETURN BENDS)

Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus O.
Carrier Corp., 302 S. Geddes St., Syracuse 1. N.Y. (*p. 25*)
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct
Chicago Nipple Mfg. Co., 1614 S. Laramie Ave., Chicago 50, Ill.
Cleveland Coppersmithing Wks., 5500 Stone Ave., Cleveland 2, O.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Dersch, Gesswein & Nevert, 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 23, N.Y. (*p. 81*)
Ellsworth Pipe & Supply Co., 513 W. Traser St., Milwaukee 3, Wis.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo
Frick Co., Waynesboro, Pa. (*p. 51*)
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 46*)
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing, Mich. (*p. 158*)
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.J.
Mueller Brass Co., Port Huron, Mich. (Copper)
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N. J.
National Pipe Bending Co., 110 River St., New Haven 13, Ct.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Engrg. Inc., 7250 E. Slauson Ave., Los Angeles 22, Cal. (*p. 196*)
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Ric-Wil Co., Union Commerce Bldg., Cleveland 14, O.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 93*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa (*p. 52*)

PIPE COVERING (See INSULATION)

PISTONS

Aluminum Industries, Inc., 2438 Beekman St., Cincinnati 25, O.
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.

PISTON PINS

Aluminum Industries, Inc., 2438 Beekman St., Cincinnati 25, O.
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Permold Co., Medina, O.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.

PISTON RINGS

American Hammered Piston Ring, Bush & Hamburg St., Baltimore, Md.
American Metallic Packing Co., 3621 Mexico St., Pittsburgh 12, Pa.
Burd Piston Ring Co., 10th & 23rd Ave., Rockford, Ill.
C. Lee Cook Mfg. Co., 916 S. 8th St., Louisville 3, Ky.
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Perfect Circle Co., 552 S. Washington, Hagerstown, Ind.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
Sealed Power Corp., 500 Sanford St., Muskegon 61, Mich.
Superior Piston Ring Co., 321 S. Cicero Ave., Chicago 44, Ill.
Wilkening Mfg. Co., 2000 S. 71st St., Phila. 42, Pa.

PITOT TUBES

E. Vernon Hill & Co., 6829 W. Highland Ave., Chicago 31, Ill

PLASTER, INSULATING (See INSULATING PLASTER & CONCRETE)

PLASTIC SHEETS, RODS & TUBES

Aetna Plywood & Veneer Co., 1799 Elston Ave., Chicago 22, Ill.
Anchor Platics Co., Inc., 533 Canal St., N.Y.C. 13
Bakelite Co., Div. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Celanese Plastics Corp., 180 Madison Ave., N.Y.C. 16
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Formica Co., 4613 Spring Grove Ave., Cin'ti 32, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Monsanto Chemical Co., Plastics Div., 600 Monsanto Ave., Springfield 2, Mass.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Premium Plastics, Inc., 627 Salem Ave., Dayton 6, O.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Sandee Mfg. Co., 5050 W. Foster Ave., Chicago 30, Ill.
Technical Ply-Woods, 228 N. La Salle St., Chicago 1, Ill.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y C. 20
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.

PLATE COILS

(A—Ammonia; B—Other refrigerants)

(B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y.
(A,B) Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. *(p. 152)*

(Continued)

GET BIG PERFORMANCE AT SMALL COST WITH

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DEAN

EVAPORATOR PLATES

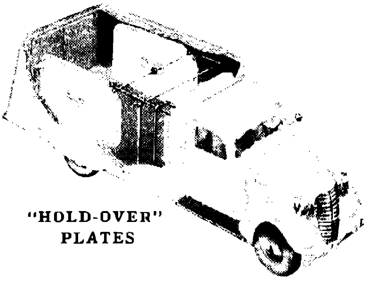
DEAN makes cold plates for ice cream cabinets, locker plants, soda fountains, farm milk coolers, liquid coolers, farm freeze cabinets, low temperature test rooms, frosted food refrigerators, window displays, food counters, refrigerated transportation and sub-zero applications for industrial chilling. Also plates for Baudelot-Type Coolers. Custom built plates available on special order such as cylinders, U's, angles, tanks, etc. Write for a copy of our latest TECHNICAL DATA BOOK.

MADE IN STAINLESS STEEL AND OTHER METALS

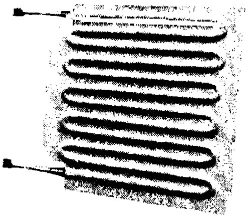
DEAN PRODUCTS, INC.

1042 Dean St. STerling 9-5400 Brooklyn 16, N.Y.

KOLD-HOLD LOW SIDES HAVE APPLICATIONS *Unlimited!*



"HOLD-OVER" PLATES



SERPENTINE PLATE

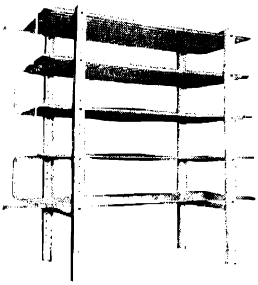
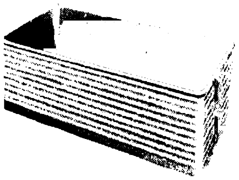


PLATE STAND



SERPENTINE LINER

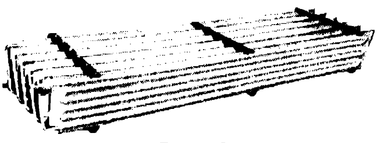


PLATE BANK

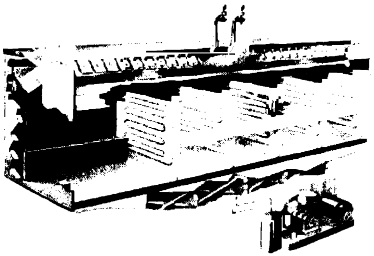
For locker plant space cooling, for shelves and stands in sharp freezing, or as cabinet liners, Kold-Hold Quick-Action Serpentine Plates, either wall mounted or in ceiling banks, have no equal in efficiency and dependability. In truck refrigeration, Kold - Hold streamlined "Hold-Over" Plates maintain the temperature of delivery truck bodies at the uniform level necessary in the successful transportation of fresh meat, ice cream and frozen foods. Specify Kold-Hold Low Sides for the most modern, efficient and economical method of refrigeration. Write today for complete data and engineering assistance.

KOLD-HOLD

Jobbers in Principal Cities

KOLD-HOLD MANUFACTURING CO.,

603 E. Hazel St., Lansing 4, Michigan



FOUNTAIN CONVERSION

PLATE COILS (Continued)

(A,B) Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
(A,B) Franklin Body & Equipment Corp., 1042 Dean St. Brooklyn 16, N.Y.
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
(A,B) Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 153)*
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(B) Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa
(B) Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago 20, Ill.

PLATE, ALLOY

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky. (Aluminum)
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm Penn Place Bldg., Pittsburgh 30, Pa.

PLATE, STEEL

Bethlehem Steel Co., Bethlehem, Pa.

HENRY

RELIEF VALVES
Approved Under Safety Codes

FREON VALVES

THE DIAPHRAGM TYPE—Operates independently of outlet pressure, not like other types of relief valves which function according to the differential between the inlet and outlet pressures. Permits the use of rupture disc assembly on outlet side without increasing initial leak pressure. Forged brass bodies, sizes ½" F.P.T., ⅝" and ⅞" O.D. solder.

STRAIGHT-THRU TYPE—Large capacity, brass construction, sizes ¼" thru 1" I.P.S. inlet; with flare or F.P.T. outlet. Positive shut-off with controlled cushion seating action.

Also other direct acting angle and straight-thru valves.

AMMONIA VALVES—Semi-steel body, sizes ⅜" thru 2" F.P.T.

WRITE FOR CATALOG

HENRY VALVE CO.
Melrose Park, Illinois, Suburb of Chicago

APPROVED FOR USE BY THE ARMY • NAVY • COAST GUARD AND THE MARITIME COMMISSION.

Central Iron & Steel Co., Harrisburg, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Lukens Steel Co., Coatesville, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Wheeling Steel Corp., Wheeling, W. Va.
Youngstown Sheet & Tube Co., Youngstown, O.

PLATING (See NAME PLATES)

PLYWOOD

Aetna Plywood & Veneer Co., 1799 Elston Ave., Chicago 22, Ill.
Technical Ply-Woods, 228 N. La Salle St., Chicago 1, Ill.

PORCELAIN, ELECTRICAL

Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Red Spot Elec. Co., Tacoma, Wash.
Westinghouse Electric Corp., East Pittsburgh, Pa.

PORCELAIN ENAMELING (See ENAMELING, PORCELAIN)

PORCELAIN ENAMEL MATERIALS

Chicago Vitreous Enamel Product Co., 1401-47 S. 55th Court, Cicero 50, Ill.
Ferro Enamel Corp., Harvard & 56th St., Cleveland 5, O.
Ingram-Richardson Mfg. Co., Beaver Falls, Pa.
Ingram-Richardson, Inc., Frankfort, Ind.
Pemco Co., 5601 Eastern Ave., Batimore 24, Md.

POROUS METAL (See POWDERED METAL PRODUCTS)

POULTRY REFRIGERATORS (See DISPLAY CASES)

POWDERED METAL PRODUCTS

Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Boston Gear Works, Quincy 71, Mass.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
International Powder Metallurgy Co., 439 W. Main St., Ridgway, Pa.
Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.
P. R. Mallory & Co., Inc., 3029 W. Washington St., Indpls., Ind.
Metals Refining Div., Glidden Co., 1717 Summer St., Hammond, Ind.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
New Jersey Zinc Co., 160 Front St., N.Y C. 17
Raybestos-Ma hattan, Inc., 61 Willett St., Passaic, N.

POWER ELEMENTS

Clifford Mfg. Co., 110 Grove St., Waltham 54, Mass.
Fulton Sylphon Co., Div., Robertshaw-Fulton Controls Co., Box 400, Knoxville 4, Tenn.

PRESSURE RELIEF DEVICES, REFRIGERANT (See also PRESSURE RELIEF VALVES)

Black, Sivalls & Bryson, Kansas City 3, Mo.
Henry Valve Co., Melrose Park, Ill. *(p. 154)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
York Corp., York, Pa. *(p. 52)*

PRESSURE RELIEF VALVES, REFRIGERANT (See also PRESSURE RELIEF DEVICES)

(A—Ammonia; B—Other refrigerants)

(A,B) Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.

Black, Sivalls & Bryson, Kansas City 3, Mo.
(A,B) A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
(A) Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 45)*
Farris Engrg. Corp., 400 Commercial Ave., Palisades Park, N.J.
(A,B) Frick Co., Waynesboro, Pa. *(p. 51)*
(A,B) Henry Valve Co., Melrose Park, Ill. *(p. 154)*
(A,B) Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 183)*
(B) Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
(B) Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
(A,B) King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 4)*
(A,B) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
J. E. Lonergan Co., 2nd & Race Sts., Philadelphia 6, Pa.
(B) Mueller Brass Co., Port Huron, Mich.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B) Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(A,B) Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
(B) Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 204)*
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

PRESSURE RELIEF VALVES, WATER

Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Climax Engrg. Co., Controls Div., 15 N. Cincinnati, Tulsa, Okla.
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.
Farris Engrg. Corp., 400 Commercial Ave., Palisades Park, N.J.
Flint & Walling Mfg. Co., Inc., 95 Oak St., Kendallville, Ind.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti 14, O.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
C. A. Norgren, 222 Santa Fe Drive, Denver, Col.
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.
Reco Products Div., Refrigeration Engrg. Corp., 202 Naudain St., Philadelphia 46, Pa.
Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sta-Rite Products, Inc., Delavan, Wis.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.

PRESSURESTATS (See CONTROLS)

PRESSURE VESSELS

Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Richard M. Armstrong Co., Box 188, West Chester, Pa.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 7
Bethlehem Steel Co., Bethlehem, Pa.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Black, Sivalls & Bryson, Kansas City 3, Mo.
Bos-Hatten, Inc., 718 E. Elk St., Buffalo, N.Y.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Dersch, Gesswein & Nevert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
John J. Dupps Co., Germantown, O.
Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Frick Co., Waynesboro, Pa. *(p. 51)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Joseph Kopperman & Sons, 312 New St., Phila. 6, Pa.
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.J.
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
McNamara & Co., Inc., 2700 Manokin St., Baltimore 30, Md.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
H. A. Phillips & Co., 3255 W. Carrol Ave., Chicago 24, Ill. *(p. 142)*
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. *(p. 70)*

(Continued)

SHANK
REFRIGERATION PRODUCTS

Ammonia Shut-Off and Line Valves, Gauge Sets, Relief Valves and Accessories

RELIEF VALVES
For Ammonia and Freon. One piece body design rust-proof finish. Internal construction stainless steel will not corrode. Positive relief action. Pressures from 75 to 400 lbs.

SEE YOUR JOBBER OR WRITE

SEMI-STEEL SHUT-OFF VALVES
Highest grade non-porous metal - full size ports - clean cut threads. Back seated stem. Shank design base perfect alignment. Long life packing ring.

CYRUS SHANK COMPANY
623 W. Jackson Blvd. Chicago 6, Ill.

PRESSURE VESSELS (Continued)

Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Scaife Co., Oakmont, Pa.
A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1, Wis.
Sta-Rite Products, Inc., Delavan, Wis.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wildman Boiler & Tank Co., 3026 Carroll Ave., Chicago 12, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

PRIMERS (See FINISHES)

PROGRAM CONTROLLERS (See CONTROLS, PROGRAM; also TIME SWITCHES)

PROPANE

Carbide & Carbon Chemicals Corp., Unit. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Esso Standard Oil Co., 15 W. 51st St., N.Y.C. 19
Standard Oil of California, 225 Bush St., San Francisco 20, Cal.
Sun Oil Co., 1608 Walnut St., Phila. 3. Pa.

PSYCHROMETERS

Bahnson Co., 1001 S. Marshall St., Winston-Salem, N.C.
Bristol Co., Waterbury 20, Ct.
Claude S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
E. Vernon Hill, 6826 W. Highland Ave., Chicago 31, Ill.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cincinnati 12, O.
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

PULLEYS

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Charles Bond Co., 617-23 Arch St., Phila., Pa.
Browning Mfg. Co., Inc., Maysville, Ky.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Congress Drive Div., Tann Corp., 3750 E. Outer Drive, Detroit 34, Mich.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
R. & J. Dick Co., Inc., 24 Sade St., Passaic, N.J.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago 24, Ill.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Mfg. Co., Lancaster, Pa.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Maurey Mfg. Corp., 2907 S. Wabash Ave., Chicago 16, Ill.
Menasha Wood Split Pulley Co., 674 Tayco St., Menasha, Wis.
Nice Ball Bearing Co. ,30th & Hunting Park Ave., Phila. 40, Pa.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Pyott Foundry & Machine Co., 328 N. Sangamon St., Chicago 7, Ill.
Rockwood Mfg. Co., 1801-2001 English Ave., Indpls. Ind.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Scientiae Corp., 101 Pine St., Dayton 2, O.
Standard-Keil Hardware Mfg. Co., Inc., 639 B'way, N.Y.C. 12
Swift Mfg. Co., Inc., 1455 E. Nine Mile Rd., Hazel Park, Mich.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*

PUMPS, AMMONIA

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. *(p. 157)*
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Cold Control, Inc., 111 Broadway, N.Y.C. 6
M. T. Davidson Co., 7 Dey St., N.Y.C. 7
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 1, N.Y.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 35)*
Le Courtenay Co., 5 Maine St., Newark 5, N.J.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
Manistee Iron Wks. Co., River St., Manistee, Mich.
Peerless Pump Div., Food Machinery & Chemical Corp., 301 West Ave., 26, Los Angeles 31, Cal.
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 25, Ill. *(p. 142)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Geo. D. Roper Corp., 340 Blackhawk Park Ave., Rockford, Ill.
Union Steam Pump Co., Battle Creek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

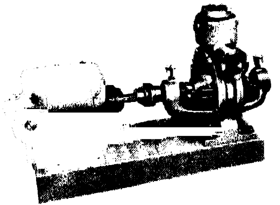
PUMPS, BRINE, WATER, etc.

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. *(p. 157)*
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. *(p. 200)*
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
Columbus Steam Pump Wks., 724 W. Gay St., Columbus 8, O.
M. T. Davidson Co., 7 Dey St., N.Y.C. 7
Dean Hill Pump Co., 4000 E. 16th St., Indpls. 7, Ind.
Decatur Pump Co., 2750 Nelson Park Rd., Decatur 78, Ill.
Deming Co., 884 S. B'way, Salem, O.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Engrg., 45 Fox St., New Haven, Ct.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Economy Faucet Co., 12 New York Ave. Newark, N.J.
Economy Pumps, Inc., Hamilton, O.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Flint & Walling Mfg. Co., Inc., 95 Oak St., Kendallville, Ind.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4

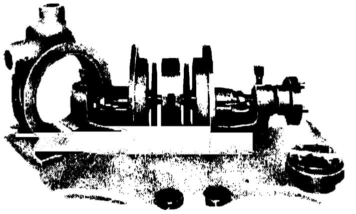
(Continued)

AURORA PUMP COMPANY

401 Loucks Street, Aurora (Chicago Suburb) Illinois
Manufacturers of
Turbine-Type, Vertical and Horizontal Centrifugal and Special Design Pumps
Distributors in Principal Cities

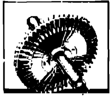

Apco Single Stage Turbine-Type Pumps.

A
COMPLETE STOCK
MAINTAINED
for
IMMEDIATE
SHIPMENT


Disassembled View Two Stage Apco Turbine-Type Pumps for high pressure duty.

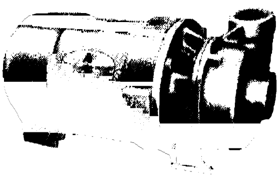
CAPACITIES UP TO 150 G.P.M.—HEADS TO 600 FT.

APCO PUMPS are IDEALLY SUITED to MANY REFRIGERATION DUTIES


The ONLY MOVING PART IN The *APCO*

FEATURES—APCO pumps are distinguished for their ability to deliver small capacities against high heads; their ability to deliver with but slight change in capacity or efficiency against drastic head variations. They possess these advanced features—SIMPLE—WEAR-FREE—COMPACT · HIGH EFFICIENCY · WILL NOT VAPOR BIND · PERFECT HYDRAULIC BALANCE · ACCESSIBILITY · HIGH SUCTION LIFT (28 ft. at sea level) · QUIET OPERATION · HIGH PRESSURE PER STAGE · DOUBLE SUCTION DESIGN · PRECISION SHAFTS · BALL BEARINGS (Support on both sides of impeller) · RIGHT OR LEFT HAND OPERATION (Changeable in Field without special parts) · REPLACEABLE COVER PLATES · AVAILABLE IN VARIOUS CORROSION RESISTANT ALLOYS.

LOW HEAD · SMALL CAPACITY—SIDE SUCTION PUMPS · MEDIUM HEAD

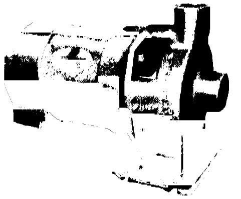

TYPE—SAC—Close Coupled
Caps. 5 to 85 G.P.M.
Heads to 45 Ft.
Sizes Available ¾", 1¼", 2"

APPLICATIONS

These pumps are ideally suited for use as integral part of Equipment Manufacturer's product, such as, air conditioning units, cooling towers, evaporator coolers, milk coolers, hot water circulators etc.; also for general service.

CONSTRUCTION SPECIFICATIONS

Made in bronze fitted, all iron or all bronze construction


TYPE—JMC—Close Coupled
Caps. 5 to 150 G.P.M.
Heads to 100 Ft.
Sizes Available 1", 1¼", 1½", 2"

CASINGS vertically split—End suction design—Made of high grade material as specified. Casing wearing ring standard construction on Type JMC.

MECHANICAL SEAL is standard equipment on all sizes and is located in packing cover—easily replaceable.

IMPELLER is balanced of enclosed type and made of high grade bronze material.

SHAFTS—Stainless Steel on Types JMC, SAC and JA. Can be furnished on Type SA at additional cost.

MOTOR built to NEMA specifications and equipped with STAINLESS STEEL shaft.

BALL BEARINGS—Permanently sealed against dust and moisture—No lubrication required.

AURORA CENTRIFUGAL PUMPS

—are available in a complete range of types and sizes to meet the manifold requirements of industry. These products of exclusive pump manufacturers are of outstanding quality in design, material and workmanship. Aurora Centrifugal Pumps are distinguished for their streamline designed impellers and waterways to insure maximum flow with least resistance and an avoidance of the turbulence which commonly causes cavitation and consequent corrosion.

WRITE
TODAY
for
CONDENSED
CATALOG "R"

PUMPS, BRINE, WATER, etc. (Continued)

Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Jacuzzi Bros., 5327 Jacuzzi Ave., Richmond, Cal.
Johnston Pump Co., 3272 E. Foothill Blvd., Pasadena 8 Calif.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Layne & Bowler, Inc., P.O. Box 6815, Hollywood Sta., Memphis 8, Tenn.
Le Courtenay Co., 15 Maine St., Newark 5, N.J.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Manistee Iron Wks. Co., River St., Manistee, Mich.
Novo Engine Co., 702 Porter St., Lansing 5, Mich.
Peerless Pump Div., Food Machinery & Chemical Corp., 301 West Ave. 26, Los Angeles 31, Cal.
Pennsylvania Pump & Compressor Co., Easton, Pa.
Piqua Machine & Mfg. Co., Young & College Sts., Piqua, O.
H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh 1, Pa.
Quimby Pump Co., Inc., Div., H. K. Porter Co., 1932 Oliver Bldg., Pittsburgh 22, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Robbins & Myers, Inc., 1345 Lagonda Ave., Springfield, O.
Geo. D. Roper Corp., 340 Blackhawk Park Ave., Rockford, Ill.
Standard Elec. Mfg. Co., W. Berlin, N.J.
Sta-Rite Products, Inc., Delavan, Wis.
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Union Steam Pump Co., Battle Creek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Wayne Home Equip. Co., 801 Glasgow Ave., Ft. Wayne 4, Ind.
Weinman Pump Mfg. Co., 290 Spruce, Columbus 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.
York Corp., York, Pa. (*p. 52*)

PUMPS, COOLANT

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. (*p. 157*)
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
Deming Co., 884 S. B'way, Salem, O.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Economy Faucet Co., 12 New York Ave., Newark, N.J.
Economy Pumps, Inc., Hamilton, O.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Jacuzzi Bros., Inc., 5327 Jacuzzi Ave., Richmond, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Manistee Iron Wks. Co., River St., Manistee, Mich.
Marlow Pumps, Ridgewood, N.J.
National Lead Co., 111 Broadway, N.Y.C. 6
Piqua Machine & Mfg. Co., Young & Coolidge Sts., Piqua, O.
H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh 1, Pa.
Quimby Pump Co., Inc., Div., H. K. Porter Co., 1932 Oliver Bldg., Pittsburgh 22, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Robbins & Myers, Inc., 1345 Lagonda Ave., Springfield, O.
Ruthman Machinery Co., 1810 Reading Rd., Cin'ti., O.
Tuthill Pump Co., 939 E. 95th St., Chicago 19, Ill.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

PUMPS, DEEP WELL

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. (*p. 157*)
A. D. Cook, Inc., Lawrenceburg, Ind.
M. T. Davidson Co., 7 Dey St., N.Y.C. 7
Decatur Pump Co. 2750 Nelson Rd., Decatur 70, Ill.
Deming Co., 884 S. Broadway, Salem, O.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Flint & Walling Mfg. Co., Inc., 95 Oak St., Kendalville, Ind.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Ingersoll Rand Co., 11 Broadway, N.Y.C. 4
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Jacuzzi Bros., Inc., 5327 Jacuzzi Ave., Richmond, Cal.
Johnston Pump Co., 2324 E. 49th St., Los Angeles 11, Cal.
Layne & Bowler, Inc., P.O. Box 6815, Hollywood Sta., Memphis 8, Tenn.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Paul Pumps, Inc., 1700 N. Harrison St., Ft. Wayne 7, Ind.
Peerless Pump Div., Food Machinery & Chemical Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Piqua Machine & Mfg. Co., Young & College Sts., Piqua, O.
A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1, Wis.
Sta-Rite Products, Inc., Delavan, Wis.
Wayne Home Equip. Co., 801 Glasgow Ave., Ft. Wayne 4, Ind.
Weinman Pump Co., 290 Spruce St., Columbus 9, O.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

PUMPS, FREON & METHYL

American-Marsh Pumps, Inc., 1948 Capital Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. (*p. 157*)
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill
Goulds Pumps, Inc., Seneca Falls, N.Y.
Le Courtenay Co., 5 Maine St., Newark 5, N.J.
Lehigh Mfg. Co., Lancaster, Pa.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
XL Refrigerating Co., 1834 W. 59th St., Chiecago 36, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

PUMPS, GEAR, VANE, ETC.

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. (*p. 157*)
Beach-Russ Co., 50 Church St., N.Y.C. 7 (*p. 159*)
Brown & Sharpe Mfg. Co., 235 Promenade St., Providence 1, R.I.
Deming Co., 884 S. Broadway, Salem, O.
Eastern Industries, Inc., 296 Elm St., New Haven, Ct.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Pumps., Inc., 50 Washington St., Brooklyn 1, N.Y.
Gray-Mills Corp., 1948 Ridge Ave., Evanston, Ill.
Goulds Pumps, Inc., 300 Fall St., Seneca Falls, N.Y.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Kinney Mfg. Co., 3529 Washington St., Boston 30, Mass. (*p. 161*)
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Peerless Pump Div., Food Machinery & Chemical Corp., 301 West Ave., 26, Los Angeles 31, Cal.
H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh 1, Pa.
Geo. D. Roper Corp., 340 Blackhawk Park Ave., Rockford, Ill.
Schutte & Koerting Co. Cornwells Heights, Bucks County, Pa.
Trimount Rotary Power Co., 189 Cedar St., Bedham, Mass.
Tuthill Pump Co., 939 E. 95th St., Chicago 19, Ill.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

PUMPS, HAND

Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Deming Co., 884 S. Broadway, Salem, O.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Flint & Walling Mfg. Co., Inc., 95 Oak St., Kendallville, Ind.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
A.Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Watson-Stillman Co., Roselle, N.J. *(p. 111)*
Worthington Pump & Machinery Corp., Harrison N.J. *(p 144)*

PUMPS, OIL

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 401 Loucks St., Aurora, Ill. *(p. 157)*
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Bowser, Inc., 1302 E. Creighton Ave., Ft. Wayne 2, Ind.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
M. T. Davidson Co., 7 Dey St., N.Y.C. 7
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Economy Faucet Co., 12 New York Ave., Newark, N.J.
Economy Pumps, Inc., Hamilton, O.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 1, N.Y.
Goulds Pumps, Inc., 300 Fall St., Seneca Falls, N.Y.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Kinney Mfg. Co., 3529 Washington St., Boston 30, Mass. *(p. 161)*
Le Courtenay Co., 5 Maine St., Newark 5, N.J.
Lunkenheimer Co., Cincinnati 14, O.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Peerless Pump Div., Food Machinery & Chemical Corp., 301 West Ave. 26, Los Angeles 31, Cal.
H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh 1, Pa.
Racine Tool & Machine Co., 1000 Carlisle Ave., Racine, Wis.
Robbins & Meyers, 1345 Lagonda Ave., Springfield, O.
Geo. D. Roper Corp., 340 Blackhawk Park Ave., Rockford. Ill.
Tuthill Pump Co., 939 E. 95th St., Chicago 19, Ill.
Union Steam Pump Co., Battrle Ceek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Watson-Stillman Co., Roselle, N.J. *(p. 111)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

PUMPS, VACUUM

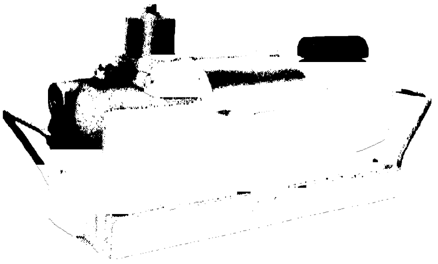
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Barrett, Haentjens & Co., Hazleton, Pa.
Beach-Russ Co., 50 Church St., N.Y.C. 7 *(p. 159)*
Bury Compressor Co., 1712 Cascade St., Erie, Pa.
Central Scientific Co., 1700 Irving Park Blvd., Chicago, Ill.
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Columbus Steam Pump Wks. Co., 724 W. Gay St., Columbus 8, O.
Croll-Reynolds Engrg. Corp., Inc., 17 John St., N.Y.C.
M. T. Davidson Co., 7 Dey St., N.Y.C. 7
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.

(Continued)

The *NEW* BEACH-RUSS

VACUUM PUMP

for Testing Air Conditioning & Refrigeration Systems



Lighter, more compact and faster, the new Beach-Russ Test Pump for air conditioning and refrigeration systems is greatly superior to the old style Beach-Russ unit which has been the favorite of the field for many years. It gives a better vacuum, and is much easier to handle and operate.

SPECIFICATIONS

Vacuum	**1/10 mm.**
Capacity	**2.5 cfm.**
Weight	**80 lbs.**
Length of Base	**28 in.**
Horse Power	**1/3 HP single end motor**
Oiling	**Automatic lubrication**

All valves eliminated.

●

Unit can be plugged in on lighting circuit. Standard motor is 110 volt, single-phase, but motor can be supplied for any required current characteristics.

Write for new bulletin on this pump

BEACH-RUSS COMPANY

Vacuum Pumps—Compressors—Gas Boosters

50 Church St. • New York 7, N.Y.

Branch Offices in Principal Cities

PUMPS, VACUUM (Continued)

Distillation Products Div., Eastman Kodak Co., Rochester 3, N.Y.
C. A. Dunham Co., 400 W. Madison St., Chicago 6, Ill.
John J. Dupps Co., Germantown, O.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Eisler Engrg. Co., Inc., 7430 S. 13th St., Newark 3, N.J.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 11, N.Y.
Fuller Co., Box 42, Catasauqua, Pa. (*p. 47*)
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Joy Mfg. Co., Oliver Bldg., Pittsburgh 22, Pa.
Kinney Mfg. Co., 3529 Washington St., Boston 30, Mass. (*p. 161*)
Leiman Bros., 100 Christie St., Newark 5, N.J.
Lummus Co., 420 Lexington Ave., N.Y.C.
Nash Engrg. Co., 358 Wilson Rd., S. Norwalk, Ct.
Pennsylvania Pump & Compressor Co., Easton, Pa.
H. K Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh 1, Pa.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
F. J. Stokes Machine Co., Tabor Rd., Phila. 20, Pa.
Union Steam Pump Co., Battle Creek, Mich.
Whittington Pump & Engrg. Corp., 1126 Prospect St., Indianapolis 3, Ind.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)

PURGERS, REFRIGERANT

Armstrong Machine Wks., 831 Maple St., Three Rivers, Mich. (*p. 160*)
Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
Belt-Ice Corp., 2845 16th Ave., W., Seattle 99, Wash. (*p. 126*)
Conoflow Corp., 2100 Arch St., Phila. 28, Pa.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Mueller Brass Co., Port Huron, Mich.
Reco Products Div., Refrigeration Engrg., Corp., 2020 Naudain St., Phila. 46, Pa.
Rex Engrg. & Sales Co., Box 4151, Oklahoma City, Okla.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

PURIFIERS, WATER (See WATER TREATING)

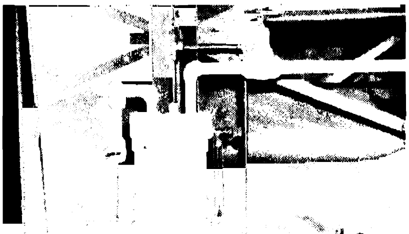
QUICK FREEZE CABINETS (See FARM & HOME FREEZERS)

QUICK FREEZERS (See FREEZERS)

RACK, GEAR & PINION

Beverage Engrg. & Equip. Co., 13301 Lakewood Hts. Blvd., Cleveland 7, O.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Ohio Gear Co., 1333 E. 179 St., Cleveland, O.
Standard Steel Specialty Co., Beaver Falls, Pa.

RACKS, WIRE (See WIRE FORMING)



ARMSTRONG
FORGED STEEL PURGERS
Save 4 Ways

Saves:
Power • Work • Refrigerant
. . . Increases Capacity

CHECK your system for excess head pressure. . . . Each 4 lbs. excess raises the load on your compressor, reduces its capacity by 1% and increases power costs 2%. Air and other non condensables collecting in the condenser and receiver are nearly always the cause. An Armstrong Purger rids the system of air and keeps it that way . . . reduces loss of refrigerant . . . saves power, labor and time. The Purger pays for itself in short order. GUARANTEED TO SATISFY.

SEND FOR BULLETIN 192 for complete details and prices.

HIGH SIDE FLOATS. Another Armstrong product for the refrigeration industry. Ask for Bulletin 1781.

ARMSTRONG MACHINE WORKS
831 Maple St., Three Rivers, Mich.

Kinney
VACUUM
PUMPS

Kinney Manufacturing Company

Boston 30, Massachusetts

NEW YORK 7, 30 Church St.
PHILADELPHIA 2, 716 Commercial Trust Bldg.
CLEVELAND 15, 2036 E. 22nd St.
LOS ANGELES 43, 3912 West Slauson Ave.
CHICAGO 3, 122 S. Michigan Ave.
SAN FRANCISCO 11, 360 Jefferson St.

KINNEY HIGH VACUUM PUMPS

Mechanical, Oil Sealed

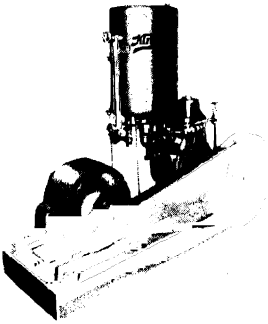
Kinney Dry Vacuum Pumps are standard equipment in the plants of the principal manufacturers of household refrigerators and quick freeze units for drying coils, compressor units and drying oil. They are also standard equipment in the plants producing temperature control units. Good mechanical performance coupled with high volumetric efficiency at the high vacuums needed to insure moisture-free and air-free units make Kinney Vacuum Pumps a necessity for such work. The same high performance is desirable for repair shops.

Kinney Vacuum Tight Valves insure tight systems.

We recommend Kinney heated separator tanks when excessive moisture is present in the system to be exhausted.

SINGLE STAGE VACUUM PUMPS

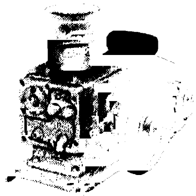
Models VSD and DVD



Eight pump sizes are available (13, 27, 46, 110, 217, 311, 486 and 702 cu. ft. per min.) with motors from ⅓ to 40 h.p. The three smallest sizes are furnished water cooled or air cooled; all larger sizes are furnished water cooled. Single Stage Pumps on a blank test will produce absolute pressure readings of 10 microns (.01 mm Hg.) or better.

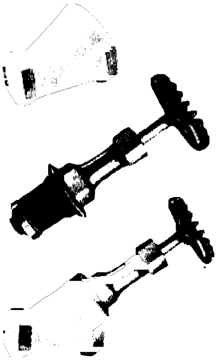
COMPOUND VACUUM PUMP

Model CVM



Four pump sizes are available (from 2 to 46 cu. ft. per min.) with ¼ to 3 h.p. motors. The working mechanism is similar to the Single Stage but the two cylinders are connected in series. The Compound Pump has an air cooled casing of unique design and special provision for oil sealing and lubricating the high vacuum working parts. Low absolute pressures to 0.5 micron (.0005 mm Hg.) are regularly maintained.

VACUUM TIGHT VALVES



Kinney Vacuum Tight Valves greatly facilitate testing installations. They are constructed with a specially designed metal bellows, eliminating the usual stuffing box. Leakage has been reduced to .000,000,007 cu. ft. per second. Available in sizes 1", 1½", 2" and 3".

Ask for Kinney Vacuum Pump Bulletin V51.

RADIATION, ULTRA-VIOLET (See LAMPS, BACTERICIDAL)

RAILROAD REFRIGERATION SYSTEMS

Cold Control, Inc., 111 Broadway, N.Y.C. 6
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. (*p. 152*)
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Propane Development Corp., 41 Murray St., N.Y.C. 7
Trane Co., La Crosse, Wis. (*p. 118*)
U. S. Thermo Control Co., 44 S. 12th St., Minneapolis, Minn.

REACH-IN REFRIGERATORS (See REFRIGERATORS, REACH-IN)

RECEIVERS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. (*p. 178*)
Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. (*p. 211*)
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Baker Refrigeration Co., S. Windham, Me. (*p. 63*)
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Bossert Co., P.O. Drawer 358, Utica 1, N.Y.
Cold Control, Inc., 111 Broadway, N.Y.C. 6
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Dean Products, Inc., 1042 Dean St., Buffalo 16, N.Y. (*p. 152*)
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32) N.Y. (*p. 31*)
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Houdaille-Hershey Corp., 1900 Foss Park Ave., North Chicago, Ill. (*p. 162*)
Industrial Wire Cloth Products Corp., Wayne, Mich. (*p. 88*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Scaife Co., Oakmont, Pa.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Standard Refrigeration Co., 332 S. Hoyne Ave., Chicago 12, Ill.
Trane Co., La Crosse, Wis. (*p. 118*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

RECEPTACLES, ELECTRICAL

Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Pent Elec. Co., Inc., 634 Michigan Trust Bldg., Grand Rapids 2, Mich.
Pyle Nat'l. Co., 1371 W. 37th St., Chicago 9. Ill.
Soreng Mfg. Corp., 9555 Eden Ave., Schiller Park, Ill.

RECIRCULATING INJECTORS (See INJECTORS)

RECORDERS, HUMIDITY

Abbeon Supply Co., 58-10 41st Dr., Woodside, N.Y.
Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Ess Instrument Co., 965 Washington Ave., Bergenfield, N.J.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Manning, Maxwell & Moore Inc., 11 Elias St., Bridgeport 2, Ct.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C 12

RECORDERS, PRESSURE

Ashton Valve Co., 43 Kendrick St., Wrentham, Mass.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Distillation Products Industries, Div. of Eastman Kodak Co., Rochester 3, N.Y.
Esterline-Angus Co., Inc., P.O. Box 596, Indpls. 6, Ind.
Fischer & Porter Co., Hatboro, Pa.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Hays Corp., Michigan City, Ind.
Instrument & Gauge Div., Electric Auto-Lite Co., Champlain St., Toledo 1, O.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti 12, O.
F. J. Stokes Machine Co., Tabor Rd., Phila. 20, Pa.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

RECORDERS, TEMPERATURE (See THERMOMETERS, RECORDING)

RECEIVERS

Hydrogen brazed, internally clean receiver tanks in sizes to meet your particular requirements. Standard diameters range from 2½" to 6" O.D. Spacing of tapped spuds and overall lengths can be varied within broad ranges.

All receivers approved by Underwriters' Laboratories.

HOUDAILLE-HERSHEY CORP.

North Chicago Division, North Chicago, Illinois

RECTIFIERS

Fansteel Metallurgical Corp., North Chicago, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indianapolis, Ind.
Selenium-Intelin Div., Federal Telephone & Radio Corp., 100 Kingsland Rd., Clifton, N.J.
United Chromium, Inc., 100 E. 42nd St., N.Y.C. 17
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

REDUCERS, SPEED (See SPEED CHANGERS)

REFRIGERANTS (See particular refrigerant)

REFRIGERATION SYSTEMS, PORTABLE

Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Blissfield Mfg. Co., 626 Depot St., Blissfield, Mich.
Dean Products Inc., 1042 Dean St., Brooklyn 16, N.Y. *(p. 152)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Philadelphia 46, Pa.
Tenney Engrg., Inc., 26 Ave. B., Newark, N.J.
York Corp., York, Pa. *(p. 52)*

REFRIGERATORS (See particular type; also DISPLAY CASES; also WALK-IN COOLERS, etc.)

REFRIGERATORS, BIOLOGICAL

Acme-National Refrigeration Co., Inc., 29-24 40th Ave., Long Island City, N.Y.
R. H. Bishop & Co., 103 N. 2nd St., Champaign, Ill.
Bowser Technical Refrigeration Div. of Bowser, Inc., Terryville, Ct.
Commercial Cooler Mfg. Co., 5736-12th St., Detroit 8, Mich.
Eimer & Amend, 633 Greenwich St., N.Y.C. 14
Evans Mfg. Corp., 460 S. 10th St., Mt. Vernon, N.Y.
Fogel Refrigerator Co., 5400 Eadom St., Philadelphia, Pa.
Foster Refrigerator Corp., Mill & N. 2nd St., Hudson, N.Y.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
Frigidaire Div., Gen'l Motors Corp., Dayton 1, O. *(p. 17)*
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Hugo Mfg. Co., 49 Ave. W. & Superior St., Duluth 10, Minn.
Jordon Refrigerator Co., 58th & Gray's Ave., Philadelphia 43, Pa.
Koch Refrigerators, 600 E. 14th Ave., N., Kansas City 16, Mo.
McCall Refrigerator Corp., Hudson, N.Y.
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
C. Schmidt Co., 1712 John St., Cincinnati 14, O.
Tenney Engrg., Inc., 26 Ave. B, Newark, N.J.
Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.

REFRIGERATORS, FLORISTS

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Commercial Cooler Mfg. Co., 5736-12th, Detroit 8, Mich
(B) Cruse Refrigerator Co., Inc., 501 W. Main St., Louisville 2, Ky.
(A,B,C) Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(A,B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
(A,B) Ed Friedrich, Inc., 1117 E. Commerce St., San Antonio 6, Tex.
(A,B,C) Gem Refrigerator Co., 165 W. Wyoming Ave., Philadelphia 40, Pa.
(A,B,C) John Herrel & Sons Co., 244 Lear St., Columbus, 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 165)*
(B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
(A,B,C) Jordon Refrigerator Co., 58th St. & Gray's Ave., Philadelphia 43, Pa.
(B) Koch Refrigerators, 600 E. 14th Ave., N., Kansas City 16, Mo.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
McCall Refrigerator Corp., Hudson, N.Y.
(B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
(A,B) Morton Show Cases, Inc., Washington Courthouse, O.
(A,B,C) John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(A) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.

(Continued)

UNIT MANUFACTURERS NOTE!

Modern Field Servicing of Critical Charged Units
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FINE CHARGING BOMBS

Tailor made to manufacturers' requirements. Capacity 6 oz. to several pounds. Assures exact field servicing or production line charging. ICC stamped for convenient shipping.

ALL FINE PRODUCTS

- ICC Refrigerant Cylinders
- Rapid Refillable Dehydrators
- Refrigerant Charging Hose
- SEALFLO Liquid Indicators
- Economy Bench Vises
- Duz-All Carbonic Gas Service Cylinders

IT'S REFILLABLE!

Fine PRODUCTS CO. 4837 SOUTH WESTERN BLVD. CHICAGO 9, ILLINOIS

REFRIGERATORS, FLORISTS (Continued)

(A,B) Simplex Mfg. Co., 1135-3rd St., Oakland 20, Cal.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Tyler Fixture Corp., 1401 Lake St., Niles, Mich. (*p. 82*)
Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B,C) Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.
(B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.

REFRIGERATORS, FOOD SERVICE (See REFRIGERATORS, REACH-IN)

REFRIGERATORS, HOUSEHOLD, ELECTRIC

Acme-Nat'l. Refrigeration Co., Inc., 29-24 40th Ave., Long Island City, N.Y.
Admiral Corp., 3800 Cortland St., Chicago 47, Ill.
Coolerator Co., Duluth 1, Minn.
Copeland Refrigeration Corp., Sidney, O. (*p. 65*)
Crosley Div., Avco Corp., Cin'ti. 25, O.
Franklin Transformer Mfg. Co., Inc., 65-22nd, N.E., Minneapolis 18, Minn.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Gibson Refrigerator Co., 515 W. Williams St., Greenville, Mich.
Hotpoint, Inc., 5600 W. Taylor St., Chicago 44, Ill.
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 5, Mich. (*p. 21*)
Leonard Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
Marvel Industries, Inc., Sturgis, Mich.
National Cooperatives, Inc., 343 S. Dearborn St., Chicago 4, Ill.
Norge Div., Borg-Warner Corp., Merchandise Mart Plaza, Chicago 54, Ill.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
Philco Corp., Tioga & C Sts., Phila. 34, Pa.
Ranney Refrigerator Co., Greenville, Mich.
Sanitary Refrigerator Co., Fond du Lac, Wis.
Universal Refrigeration Co., 5001 W. Century Blvd., Inglewood, Cal.
Westinghouse Elec. Corp., Mansfield, O.

REFRIGERATORS, HOUSEHOLD, ABSORPTION

Rheem Mfg. Co., 570 Lexington Ave., N.Y.C. 22
Servel, Inc., Evansville 20, Ind.

REFRIGERATORS, HOUSEHOLD, ICE

American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Coolerator Co., 50 Ave. W. and Wadena St., Duluth 1, Minn.
Ice Cooling Appliance Corp., Morrison, Ill.
Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
Sanitary Refrigerator Co., Fond du Lac, Wis.

REFRIGERATORS, HOUSEHOLD, PRIVATE LABEL

Artkraft Mfg. Co., E. Kibbey St., Lima, O.
Franklin Transformer Mfg. Co., Inc., 65-22nd., N. E., Minneapolis 18, Minn.
Ranney Refrigerator Co., Greenville, Mich.

REFRIGERATORS, LOW TEMPERATURE FOR LABORATORY & INDUSTRIAL USE

Aminco Refrigeration Products Co., 14544 3rd Ave., Detroit 3, Mich. (*p. 149*)
R. H. Bishop & Co., 103 N. 2nd St., Champaign, Ill.
Bowser Technical Refrigeration Div. of Bowser, Inc., Terryville, Ct.
Brewer-Titchener Corp., Binghamton, N.Y.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Chapman & Wood Refrigeration, 4825 Southwest Pomona, Portland 19, Ore.
Eimer & Amend, 633 Greenwich St., N.Y.C. 14
Evans Mfg. Corp , 460 S. 10th St., Mt. Vernon, N.Y.
Fogel Refrigerator Co., 5400 Eadom St., Philadelphia, Pa.
Foster Refrigerator Corp., Mill & N. 2nd St., Hudson, N.Y.
Freezer Box Div., Annapolis Yacht Yard, Box 791 Annapolis, Md. (*p. 207*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gem Refrigerator Co., 165 W. Wyoming Ave., Philadelphia 40, Pa.
John Herrel & Sons, Inc., 244 Lear St., Columbus 6, O.
Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. (*p. 165*)
Hussman Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 88*)
Jordon Refrigerator Co., 58th St., & Grays Ave., Philadelphia 43, Pa.
Koch Refrigerators, 600 E. 14th Ave., N., Kansas City, 16, Mo.
McCray Refrigerator Co., Kendallville, Ind.
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Philadelphia 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
C. Schmidt Co., 1712 John St., Cincinnati 14, O.
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Sweden Freezer Mfg. Co., 3401-17th Ave., West, Seattle 99, Wash.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 51, N.J.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
Wilson Refrigeration, Inc., S. Main St., Smyrna, Del.
York Corp., York, Pa. (*p. 52*)

REFRIGERATORS, MORTUARY

Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. (*p. 165*)
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.

REFRIGERATORS, REACH-IN

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit; D—Ice refrigerated)

Bally Case & Cooler Co., Bally, Pa.
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
(A,B,D) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(A,B,C,D) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(C) Freezer Box Div., Annapolis Yacht Yard, Box 791, Annapolis, Md. (*p. 207*)
(A,B,D) Ed Friedrich, Inc., 1117 E. Commerce St., San Antonio 6, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(A,B,C,D) Gem Refrigerator Co., 165, W. Wyoming Ave., Phila. 40, Pa.
(A) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. (*p. 29*)
Harder Refrigerator Div., Tyler Fixture Corp., Cobleskill, N.Y.
(A,B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.

(A,B,C,D) **Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia.** *(p. 165)*

(A,B) C. V. Hill & Col, Inc.,360 Pennington Ave., Trenton 1, N.J.

(A) **Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo.** *(p. 88)*

(A,B,C,D) Jordon Refrigerator Co., 58th St. & Grays Ave., Phila. 43, Pa.

Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*

(A,B) Koch Refrigerators, 600 E. 14th St., North Kansas City 16, Mo.

(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.

(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.

(A,B) McCray Refrigerator Co., Kendallville, Ind.

(A,B) Masterfreeze Corp., Sister Bay, Wis.

Matthews Refrigerator & Door Co., 5103 S.E. Powell Blvd., Portland 6, Ore.

(A,B) Morton Show Cases, Inc., Washington Courthouse, O.

(A,B,C) John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.

(A,B) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.

(A,B,C,D) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.

(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.

C. L. Percival Refrigerator Co., 1805 E. 4th St., Boone, Ia.

(A) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.

(A,B,C) J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.

(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.

Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 20, Pa.

(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.

(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.

(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.

(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.

(B) Southern Fixture Mfg. Co., P.O. Box 245, Greensboro N.C.

(B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.

Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.

(B) **Tyler Fixture Corp., 1401 Lake St., Niles, Mich.** *(p. 82)*

(A,C) United Refrigerator Co., Hudson, Wis.

Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.

Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.

(A,B) Warren Co., Inc., P.O. Box 1435, Atlanta 1, Ga.

(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.

(A,B,C) Wilson Refrigeration, Inc., S. Main St., Smyrna, Del.

REFRIGERATORS, RESTAURANT (See REFRIGERATORS, REACH-IN)

REFRIGERATORS—SALAD COUNTERS

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.

Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. *(p. 152)*

Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.

Fogel Refrigerator Co., 5400 Eadom St., Philadelphia, Pa.

Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.

John Herrel & Sons Inc., 244 Lear St., Columbus 6, O.

C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.

Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.

John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.

Stainless Food Equip. Co., 272 New St., Newark, N.J.

Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.

Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.

REGISTERS (See GRILLES)

REGULATORS (See particular type following; also CONTROLS; also CONTROLLERS)

REGULATORS, CARBON DIOXIDE

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.

Conoflow Corp., 2100 Arch St., Phila. 28, Pa.

Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.

Dockson Corp., 3839 Wabash Ave., Detroit 8, Mich.

Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.

Hays Corp., Michigan City, Ind.

Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.

C. A. Norgren, 222 Santa Fe Drive, Denver, Col.

Pure Carbonic Co., Div. of Air Reduction Co., 60 E. 42nd St., N.Y.C. 17

Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.

REGULATORS, FLOW

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.

Barber-Colman Co., Rockford, Ill.

Bristol Co., Waterbury 20, Ct.

Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.

Climax Controls Div., Black, Sivalls & Bryson, Inc., Box 1529, Tulsa 1, Okla.

Conoflow Corp., 2100 Arch St., Phila. 28, Pa.

Davis Regulator Co., 2541 S. Washtenaw Ave., Chicago 8, Ill.

Fischer & Porter Co., Hatboro, Pa.

Foxboro Co., 38 Neponset Ave., Foxboro, Mass.

General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 64)*

Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.

(Continued)

HERRICK STAINLESS STEEL REFRIGERATORS

Standard and Custom Built
Reach-Ins . . . Step-Ins . . . Walk-Ins

Performance-Proved for Hotels, Restaurants, Clubs, Hospitals, Universities, Colleges, Institutions, School and Factory Cafeterias.

MODEL R5566

WRITE FOR INFORMATION
HERRICK REFRIGERATOR COMPANY
1019 COMMERCIAL ST., WATERLOO, IOWA

REGULATORS, FLOW (Continued)

Hays Corp., Michigan City, Ind.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Klipfel Valves, Inc., 1000 Weller Ave., Hamilton, O.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Refrigerating Specialties Co., 728 S. Sangamon Blvd., Chicago 12, Ill.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Trane Co., La Crosse, Wis. *(p. 118)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.

REGULATORS, LIQUID LEVEL (See CONTROLS LIQUID LEVEL)

REGULATORS, PRESSURE, TEMPERATURE, etc.

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Climax Controls Div., Black, Sivalls & Bryson, Inc., Box 1529, Tulsa 1, Okla.
Conoflow Corp., 2100 Arch St., Phila. 28, Pa.
Coral Designs, 12-16 Astoria Blvd., Long Island 2, N.Y.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 40)*
Dockson Corp., 3839 Wabash Ave., Detroit 8, Mich.
Electric Sprayit Co., 1415 Illinois Ave., Sheboygan, Wis.
Erickson Specialty Co., 10 Cayuga St., Cohoes, N.Y.
Fischer & Porter Co., Hatboro, Pa.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Fulton Sylphon Division, Robertshaw-Fulton Controls Co., Knoxville 4, Ten.
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11, N.J.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 64)*
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 83)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Klipfel Valves, Inc., 1000 Weller Ave., Hamilton, O.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport, Ct.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 7, Minn. *(p. 67)*
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
C. A. Norgren, 222 Santa Fe Drive, Denver, Colo.
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Pittsburgh Equitable Meter Div., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Sta-Rite Products, Inc., Delavan, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.
Wheelco Instruments Co., Harrison & Peoria Sts., Chicago 7, Ill.
White-Rodgers Electric Co., 1209 Cass Ave., St. Louis 6, Mo.
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

REGULATORS, VOLTAGE

Acme Elec. Corp., Cuba, N.Y.
American Transformer Co., 172 Emmet St., Newark 5, N.J.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Sola Elec. Co., 4633 W. 16th St., Chicago 50, Ill.
Superior Electric Co., Bristol, Ct.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

RELAYS, ELECTRICAL (See also STARTERS)

Allen-Bradley Co., Milwaukee 4, Wis.
American Relay & Controls, Inc., 4901 Flournoy St., Chicago 44, Ill.
Automatic Elec. Mfg. Co., 10 State St., Mankato, Minn.
Automatic Switch Co., 391 Lakeside Ave., Orange, N.J.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
R. W. Cramer Co., P.O. Box 25, Centerbrook, Ct.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
DuraKool, Inc., 1010 Main St., Elkhart, Ind.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 64)*
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
H-B Instrument Co., 2646 Trenton Ave., Philadelphia 25, Pa.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Roller-Smith Div., Realty & Industrial Corp., Bethlehem, Pa.
Signal Engrg. & Mfg. Co., 154 W. 14th St., N.Y.C. 11
Spencer Thermostat Div. of Metals & Controls Corp. 34 Forest St., Attleboro, Mass.
Square D Co., Inc., 6060 Rivard St., Detroit 11, Mich.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
White-Rodgers Electric Co., 1209 Cass Ave., St. Louis 6, Mo.

RELIEF VALVES (See PRESSURE RELIEF VALVES)

Sarco Company, Inc.

Branches in Principal Cities

Empire State Building New York 1, N.Y.

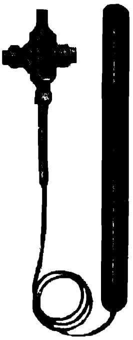
SARCO CANADA LIMITED, 496 Church St., Toronto 5, Ont., Canada

AIR CONDITIONING ACCESSORIES

Sarco products have kept pace with the rapid developments in Air Conditioning. Two types of temperature control, blenders for controlling brine flow, steam traps and strainers are available in a complete range of sizes.

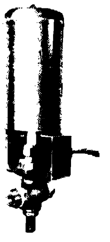
SELF-CONTAINED TEMPERATURE REGULATORS

Type TR-21


Type KR-14

For ducts, reheaters, drinking water control, unit coolers and brine coils. They are simple, self-operated valves—using the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. A type and size for every purpose—for steam, gas, oil, water or brine, for temperatures ranging from 0 to 300° F. Catalog No. RD600.

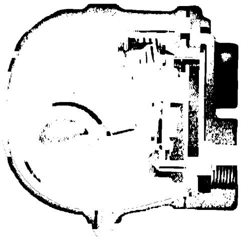
ELECTRIC CONTROLS


Motor Valve

Thermostat

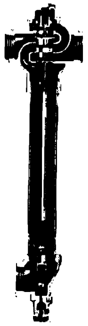
For refrigeration and air conditioning. When temperature control must be close, when wide and rapid fluctuations must be corrected instantly, when the thermostat is at a distance from the controlled unit, use electric regulators. They consist of room thermostats, aquastats, and limit controls for all heating and air conditioning needs; also motor valves for steam, water or brine. Catalog No. RD1000.

FLOAT-THERMOSTATIC TRAPS



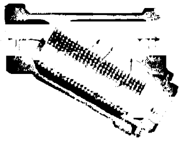
For dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ to 2 in. Catalog No. RD450.

BLENDERS



For mixing warm and cold brine or water for cooling coils and unit coolers. Also mix very hot and cold water and deliver blended water at any temperature for which they have been adjusted. Valves are fully balanced so that control is not disturbed by differing or fluctuating water or brine pressures. Available in sizes ½ in. to 4 in., for pressures up to 150 lb. Catalog Nos. RD700 and RD800.

SELF-CLEANING STRAINERS



For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 500 psi with brass, iron or monel screens. Available in sizes ¼ to 8 in. Catalog No. RD1200.

RESISTORS, ELECTRICAL

Carborundum Co., P.O. Box 337, Niagara Falls, N.Y.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Clarostat Mfg. Co., Inc., 130 Clinton St., Brooklyn 2, N.Y.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
I-T-E Circuit Breaker Co., 19th & Hamilton Sts., Phila. 30, Pa.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls. Ind.
Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Ohmite Mfg. Co., 4835 W. Flournoy St., Chicago 44, Ill.
Rostone Corp., Lafayette, Ind.
Sprague Products Co., Marshall St., N. Adams, Mass.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Edwin L. Wiegand Co., 7506 Thomas Blvd., Pittsburgh 8, Pa.

RESPIRATORS (See GAS MASKS)

RETARDERS, CORROSION (See INHIBITORS)

RETARDERS, DOUGH (See DOUGH RETARDERS)

RETURN BENDS

Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Fitzsimons Mfg., 3775 E. Outer Dr., Detroit 12, Mich.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Henry Valve Co., Melrose Park, Ill. (*p. 104*)
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
C. F. Moores Co., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
Mueller Brass Co., Port Huron, Mich.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Taylor Forge & Pipe Wks., P.O. Box 485, Chicago 90, Ill.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17

REVOLUTION COUNTERS (See COUNTING & COMPUTING DEVICES)

RHEOSTATS

Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Ohmite Mfg. Co., 4835 W. Flournoy St., Chicago 44, Ill
O. A. Sutton Corp., 1812 W. 2nd St., Wichita 1, Kan.
Vulcan Elec. Co., 88 Holton St., Danvers, Mass.
Westinghouse Elec. Co., 4454 Genesee St., Buffalo 5, N.Y

RINGS, PIPE WELDING

All-State Welding Alloys Co., Inc., 273 Ferris Ave., White Plains, N.Y.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

RIVET TREATING CABINETS (See REFRIGERATORS, LOW TEMPERATURE)

RIVETS

Aluminum Co. of America, Pittsburgh 19, Pa.
W. Ames & Co., Jersey City, N.J.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cherry Rivet Co., 231 Winston St., Los Angeles 13, Cal.
Clark Bros. Bolt Co., Milldale, Ct.
Crescent Belt Fastener Co., 480 Lexington Ave., N.Y.C. 17
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. (Explosive)
Fansteel Metallurgical Corp., North Chicago, Ill. (Contact)
General Plate Div., Metals & Controls Corp., Attleboro, Mass. (Precious Metals)
H. M. Harper Co., 8200 Lehigh Ave., Morton Grove, Ill.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburgh 19, Pa. (Copper)
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind. (Precious Metals)
Oliver Steel Corp., S. 10th & Muriel Sts., Pittsburgh 3, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
South Chester Corp., 1418 S. Penn Sq., Philadelphia 2, Pa. (*p. 101*)
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Townsend Co., 205 River Rd., New Brighton, Pa

ROD (See BAR & ROD)

ROD, DRILL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Bethlehem Steel Co., Bethlehem, Pa.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Vanadium-Alloys Steel Co., Latrobe, Pa.

ROD, THREADED & TIE

Allmetal Screw Products Co., Inc., 33 Greene St., N.Y.C. 13
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
Eastern Machine Screw Corp., Truman & Barclay Sts., New Haven, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 51, Mo.

ROOF COOLERS & ROOF COOLING SYSTEMS

April Showers Co., 4126-8th St., N.W., Washington 11, D.C. (*p. 169*)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Siegfried Ruppright, Box 6795, Los Angeles 22, Cal.
Spraying Systems Co., 3201 Randolph St., Bellwood, Chicago, Ill.
Water Cooling Corp., 71Nassau St., N.Y.C. 7
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo.

ROOM COOLERS (See AIR CONDITIONERS)

RUBBER, SYNTHETIC, & RUBBER-LIKE PLASTICS

Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Bakelite Co., Div. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Canfield Rubber Co., 708 Railroad Ave., Bridgeport 5, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
Dennis Chemical Co., 2701 Papin St., St. Louis 3, Mo.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Golden State Rubber & Latex Corp., 657 E. 61st St., Los Angeles, Cal.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber Co., Inc., Box 417, Goshen, Ind.
Haveg Corp., Marshallton, Del.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Inland Mfg. Div., Gen'l Motors Corp., Dayton 1, O.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Rubatex Div., Great American Industries, Bedford, Va. (p. *130*)
Thermoid Co., Trenton, N.J.
Amos Thompson Corp., Edinburgh, Ind.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.
S. S. White Dental Mfg. Co., 10 E. 40th St., N.Y.C.

RUBBER PRODUCTS, CELLULAR

Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
Dryden Rubber Div., Sheller Mfg. Corp., 1014 S. Kildare Ave., Chicago 24, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 85)*
Robinson Aviation, Inc., Teterboro, N.J.
Rubatex Div., Great American Industries, Bedford, Va. *(p. 130)*
Sponge Rubber Products Co., 1600 Derby Place, Shelton, Ct.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

RUBBER PRODUCTS, EXTRUDED

Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Ball Bros. Co., Muncie, Ind.
Canfield Rubber Co., 708 Railroad Ave., Bridgeport 5, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
Dryden Rubber Div., Sheller Mfg. Corp., 1014 S. Kildare Ave., Chicago 24, Ill.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 171)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 85)*
Judsen Rubber Works, Inc., 4107 West Kinzie, Chicago 24, Ill.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 8, N.J.
Rubatex Div., Great American Industries, Bedford, Va. *(p. 130)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

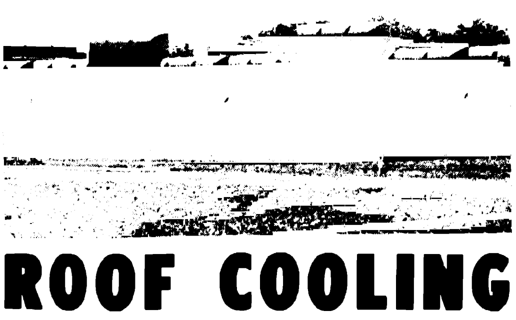
RUBBER PRODUCTS, HARD

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Judsen Rubber Works, Inc., 4107 West Kinzie, Chicago 24, Ill.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
Luzerne Rubber Co., Trenton 9, N.J.
Rubatex Div., Great American Industries, Bedford, Va. *(p. 130)*
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
Thermoid Co., Trenton, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

RUBBER PRODUCTS, MECHANICAL BONDED

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Bushings, Inc., 4358 Coolidge Highwy., Royal Oak, Mich.

(Continued)



- Reduces Air Conditioning Loads as much as 25%.
- Lowers Temperatures of Upper Floors 10° to 15°
- For Industrial, Commercial and Residential Bldgs.
- Used on all Types and Shapes of Roofs.
- Installation is Simple, Inexpensive and Foolproof.
- Operating and Maintenance Costs are Negligible.
- Thermostatically Controlled; no Pool, no Runoff.
- Millions of Sq. Ft. of roofs are cooled by A.S.*

Besides cooling "under-roof" areas, April Showers adds longer life to the roof, gives added protection in case of fires and acts as a lightning arrestor. WRITE for detailed, technical data.

* April Showers are patented.

ROOF COOLING
Spray Method

APRIL SHOWERS COMPANY, INC.
4126 Eighth Street, N.W., Washington 11, D.C.

RUBBER PRODUCTS, MECHANICAL BONDED (Continued)

Dryden Rubber Div., Sheller Mfg. Corp., 1014 S. Kildare Ave., Chicago 24, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
Golden State Rubber & Latex Corp., 657 E. 61st St., Los Angeles, Cal.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
Quaker Rubber Corp., Tacony & Comly Sts., Philadelphia 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor & Webster St., Trenton 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

RUBBER PRODUCTS, MOLDED

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Apex Molded Products Co., 3574 Ruth St. 34, Phila. Pa.,
Ball Bros. Co., Muncie, Ind.
Canfield Rubber Co., 708 Railroad Ave., Bridgeport 5, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
Cornish Wire Co., Rutland, Vt.
Darcoid Co., Inc., 145-6th Ave., N.Y.C. 13
Dryden Rubber Div., Sheller Mfg. Corp., 1014 S. Kildare Ave., Chicago 24, Ill.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Garlock Packing Co., 402 E. Main St., Palymra, N.Y.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
Golden State Rubber & Latex Corp., 657 E. 61st St., Los Angeles, Cal.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Julsen Rubber Works, Inc., 4107 West Kinzie, Chicago 24, Ill.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Robinson Aviation, Inc., Teterboro, N.J.
Rubatex Div., Great American Industries, Bedford, Va. (*p. 130*)
Sponge Rubber Products Co., 1600 Derby Place, Shelton, Ct.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
Thermoid Co., Trenton, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

RUPTURE MEMBERS (See PRESSURE RELIEF DEVICES)

SADDLES, PIPE (See FITTINGS; also HANGERS, PIPE)

SAFETY SHUT-OFFS FOR GAS REFRIGERATORS

Spencer Thermostat Div., Metals & Controls Corp. 34 Forest St., Attleboro, Mass.

SAFETY VALVES (See PRESSURE RELIEF VALVES)

SALAD REFRIGERATORS (See REFRIGERATORS, SALAD)

SCALE INHIBITORS (See WATER TREATING OR COATINGS & COMPOUNDS)

SCALE TRAPS (See also STRAINERS; also FILTERS, etc.)

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Dersch, Gesswein & Nevert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. (*p. 46*)
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. (*p. 102*)
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. (*p. 183*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (p. *89*)
Neptune Meter Co., 50 W. 50th St., N.Y.C. 20
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. (*p. 142*)
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. (*p. 155*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 88*)
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p. 52*)

SCREENS, PERFORATED METAL (See METAL, PERFORATED)

SCREENS, VIBRATING

Cambridge Wire Cloth Co., Cambridge, Md.
Deister Concentrator Co., 901 Glasgow Ave., Ft. Wayne Ind.
Hendrick Mfg. Co., Carbondale, Pa.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Kentucky Metal Products Co., Preston St., & Audubon Pk., Louisville 13, Ky.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
D. V. Murray Mfg. Co., 1024-3rd St., Wausau, Wis.
Read Machinery Div., Standard Stoker Co., Inc., York, Pa.
Robins Conveyors Div., Hewitt-Robins, Inc., Passaic, N.J.

SCREENS, WIRE (See WIRE CLOTH)

SCREW MACHINE PARTS

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Corbin Screw Corp., New Britain, Ct.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Eastern Machine Screw Corp., Truman & Barclay Sts., New Haven 6, Ct.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Mueller Brass Co., Port Huron, Mich.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Townsend Co., 205 River Rd., New Brighton, Pa

SCREWS (See BOLTS)

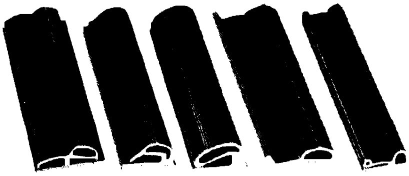
THE GENERAL TIRE & RUBBER CO.

MAKERS OF AMERICA'S TOP QUALITY TIRE

Mechanical Goods Division • Wabash, Indiana

GENERAL *Engineered Rubber Parts*

Rubber parts for refrigerating and air-conditioning equipment, made to your specification and General quality standards. General engineers offer skilled help in selecting the right rubber and design for top efficiency.



RUBBER DOOR GASKETS

Extruded rubber gaskets in any design and size, solid or hollow, made accurately to specification. Natural or synthetic rubber. Our method of compounding produces gaskets that are odorless, tasteless and non-staining, with excellent compressibility. Shown above are gaskets supplied by us to leading refrigerator makers.

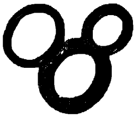
BONDED-TO-METAL PARTS

General is experienced in making rubber-bonded-to-metal parts of all kinds—hard and soft rubber knobs, cooler lids, fountain fittings, bumpers, and other parts.

MOLDED RUBBER PARTS

General is equipped to supply all types of molded rubber parts in any size and design, made to specification, of hard and soft rubber. Dust seal and rubber bushing shown are typical of many refrigerator parts molded by General.

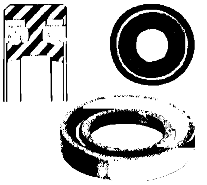
DIE-CUT RUBBER PARTS



General is a reliable source of supply for die-cut rubber parts—washers, gaskets, etc. Made to your specification, to close tolerances.

OIL AND REFRIGERANT SEALS

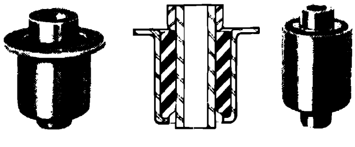
Shown are single and double-lip oil seals designed by General for highest efficiency and service. General engineers can help you design seals for lubricants and refrigerants to meet your special needs.



Precision Vibration Control with

GENERAL SILENTBLOC MOUNTINGS

GENERAL Silentbloc Mountings offer precision control of vibration and noise in motors, compressors, fans and pumps. Engineered to conform to any curve of deflection, they give complete control of vibration through starting and operating cycles. Silentbloc performance is due to its patented construction—elongation and confinement of rubber between metal sleeves. Radial compressive force, with even distribution of stress, assures indestructible rubber-to-metal adhesion and long service.



Silentbloc Mountings can solve your problems of vibration and noise in domestic and commercial equipment. Made of any metal, in any size to carry loads of ounces to tons. Simple to incorporate in your designs, easy to install. Write for Silentbloc booklet.

SCROLLS, BLOWER (See BLOWERS, WHEELS, HOUSINGS)

SCUTTLE BUTTS (See DRINKING WATER COOLERS)

SEALING COMPOUNDS (See COMPOUNDS)

SEALS, DOOR BOTTOM (See also DOOR GASKETS)

Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
W. J. Dennis & Co., 1732 N. Kolmar Ave., Chicago 39, Ill.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. (*p. 85*)
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 (*p. 136*)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
York Corp., York, Pa. (*p. 52*)

SEALS, OIL, SHAFT, etc. (See also GASKETS, etc.)

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Cartriseal Corp., 1450 W. Randolph St., Chicago 7, Ill.
A. W. Chesterton Co., 4 Ashland St., Everett 49, Mass.
Durametallic Corp., 2104 Factory St., Kalamazoo 24F, Mich.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 135*)
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
National Motor Bearing Co., Inc., Redwood City, Cal.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Speer Carbon Co., St. Marys, Pa.
U. S. Graphite Co., Saginaw, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.

SEALS, SHAFT, MECHANICAL

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Bridgeport Thermostat Div. of Robertshaw-Fulton Controls Co., 1225 Connecticut Ave., Bridgeport 1, Ct.
Chicago Seal Co., 332 S. Hoyne Ave., Chicago 20, Ill.
Clifford Mfg. Co., 110 Grove St., Waltham 54, Mass.
Crane Packing Co., 1800 Cuyler Ave., Chicago 13, Ill.
Rotary Seal Co., 2020 N. Larrabee St., Chicago 14, Ill.

SEAMLESS TUBING (See TUBES & TUBING)

SHAFTS
Machined to your specification on a production basis. Send us your prints. Specialized in this work.
MODERN MACHINE WORKS INC.
5350 S. Kirkwood Ave.
CUDAHY, WIS.

SEATS, COMPRESSOR VALVE

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Chicago Seal Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.

SELF-SERVICE REFRIGERATORS (See DISPLAY CASES)

SEPARATORS (See particular type, i.e., OIL; also STEAM, etc.)

SHAFTS & SHAFTING (See also CRANKSHAFTS)

Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Modern Machine Works, Inc., 5350 S. Kirkwood Ave., Cudahy, Wis. (*p. 172*)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Spicer Mfg. Co., 4100 Bennett Rd., Toledo 1, O.

SHAFT COUPLINGS (See also FLEXIBLE COUPLINGS)

Ajax Flexible Coupling Co., Inc., Westfield, N.Y.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American Flexible Coupling Co., 1808 Pittsburgh Ave., Erie, Pa.
Charles Bond Co., 617-23 Arch St., Phila., Pa.
Bond Foundry & Machine Co., Manhem, Pa.
Boston Gear Works, Quincy 71, Mass.
Bushings, Inc., 4358 Coolidge Highwy.,Royal, Oak, Mich
Capitol Mfg. & Supply Co., 153 W Fulton St., Columbus, O.
Centric Clutch Co., Woodbridge, N.J.
Chicago Die Casting Mfg.Co., 2500 W. Monroe St., Chicago 12, Ill.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Congress Drive Div., Tann Corp., 3750 E. Outer Drive, Detroit 34, Mich.
Continental Diamond Fibre Co., 3 Chapel St., Newark, Del.
Diamond Chain Co., Inc., 402 Kentucky Ave., Indpls. 7, Ind.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
W. A. Jones Foundry & Machine Co., 4401 Roosevelt Rd., Chicago 24, Ill.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.
Simonds Gear & Mfg. Co., 2501 Liberty Ave., Pittsburgh 22, Pa.
Standard Pressed Steel Co., Box 824, Jenkintown, Pa.
Twin Disc Clutch Co., Racine, Wis.

SHAFT SEALS (See SEALS)

SHAPES, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Brasco Mfg. Co., Harvey, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Melrath Supply & Gasket Co., Inc., Tioga & Memphis Sts., Phila. 34, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.

SHAPES, BRASS, BRONZE & COPPER

American Brass Co., Waterbury 88, Ct. *(p. 189)*
Bohn Aluminum R Brass Corp, E. Maumee, Adrian, Mich.
Brasco Mfg. Co., Harvey, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Titan Metal Mfg. Co., Bellefonte, Pa.

SHAPES, DRAWN OR PRESSED (See Also STAMPINGS; also SHELLS)

Ackermann Mfg. Co., Wheeling, W. Va.
Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 178)*
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Commercial Shearing & Stamping Co., 1775 Logan St., Youngstown, O.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Greene Mfg. Co., Inc., 1028 Douglas Ave., Racine, Wis.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Lansing Stamping Co., 1159 S. Pennsylvania Ave., Lansing 2, Mich.
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Powell Pressed Steel Co., Hubbard, O.
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. *(p. 79)*
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Stanley Wks., 195 Lake St., New Britain, Ct.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O

SHAPES, STRUCTURAL

Aluminum Co. of America, Pittsburgh 19, Pa.
Atlantic Steel Co., P.O. Box 1714, Atlanta, Ga.
Bethlehem Steel Co., Bethlehem, Pa.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Brasco Mfg. Co., Harvey, Ill.

Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Weirton Steel Co., Weirton, W. Va.

SHAPES, VULCANIZED FIBRE (See FIBRE)

SHARP FREEZERS (See FREEZERS)

SHEAVES (See PULLEYS)

SHEET, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Fairmount Aluminum Co., Fairmount, W. Va.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
Rigidized Metals Corp., 658 Ohio St., Buffalo 3, N.Y.
United Smelting & Aluminum Co., Inc., New Haven, Ct.

SHEET, ASBESTOS

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Belmont Packing & Rubber Co., Butler & Sepviva Sts., Phila. 7, Pa.
Philip Carey Mfg. Co., Lockland Cin'ti. 15, O.
Dareoid Co., Inc., 145-6th Ave., N.Y.C. 13
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
Hollow Center Packing Co., 6523 Euclid Ave., Cleveland 3, O.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Keasbey & Mattison Co., Ambler, Pa.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Linear, Inc., State Rd., & Levick St., Phila. 35, Pa.
Melrath Supply & Gasket Co., Inc., Tioga & Memphis Sts., Phila. 34, Pa.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Norristown Magnesia & Asbestos Co., Washington St. below Ford St., Norristown, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Rutland Fire Clay Co., Curtis Ave., Rutland, Vt.
Standard Asbestos Mfg. Co., 860 W. Evergreen Ave., Chicago 22, Ill.
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
Grant Wilson, Inc., 316 S. La Salle St., Chicago 4, Ill.

SHEET, BRASS, BRONZE, COPPER

American Brass Co., Waterbury, Ct. *(p. 189)*
Berger Mfg. Div. of Republic Steel Corp., 1038 Belden Ave., N.E., Canton 5, O.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
C. G. Hussey & Co., 2860-2nd Ave., Pittsburgh 19, Pa.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indianapolis, Ind.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
SisalKraft Co., 205 W. Wacker Drive, Chicago 6, Ill.
Western Brass Mills, Div. of Olin Industries, Inc., E. Alton, Ill.

SHEET, ENAMELING

Armco Steel Corp., Middletown, O.
Berger Mfg. Div., Republic Steel Corp., 1038 Belden Ave., N.E., Canton 5, O.
Central Steel & Wire Co., 300 W. 51st St., Chicago 32, Ill.
Inland Steel Co., 38 S. Dearborn St., Chicago 13, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Rigidized Metals Corp., 658 Ohio St., Buffalo 3, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Weirton Steel Co., Weirton, W.Va.

SHEET, INCONEL, MONEL & NICKEL

International Nickel Co., 67 Wall St., N.Y.C. 5

SHEET, STEEL (See also particular type)

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Armco Steel Corp., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Inland Steel Co., 38 S. Dearborn St., Chicago 13, Ill.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Reeves Steel & Mfg. Co., 137 E. Iron Ave., Dover, O.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Wallingford Steel Co., Wallingford, Ct.
Weirton Steel Co., Weirton, W. Va.
Wheeling Corrugating Co., Wheeling, W. Va.
Wheeling Steel Corp., Wheeling, W. Va.
Youngstown Sheet & Tube Co., Youngstown, O.

SHEET, STEEL, COATED OR GALVANIZED

Armco Steel Corp., Middletown, O.
Berger Mfg. Div., Republic Steel Corp., 1038 Belden Ave., N.E., Canton 5, O.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
A. B. Murray Co., Inc., 630 Greene Lane, Elizabeth, N.J.
Reeves Steel & Mfg. Co., 137 E. Iron Ave., Dover, O.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Weirton Steel Co., Weirton, W. Va.
Wheeling Corrugating Co., Wheeling, W. Va.
Wheeling Steel Corp., Wheeling, W. Va.

SHEET, STEEL, ELECTRICAL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Armco Steel Corp., Middletown, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Wheeling Steel Corp., Wheeling, W. Va.

SHEET, STEEL, STAINLESS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Armco Steel Corp., Middletown, O.
Berger Mfg. Div., Republic Steel Corp., 1038 Belden Ave., N.E., Canton 5, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Eastern Stainless Steel Corp., P.O. Box 1975, Baltimore 3, Md.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Rigidized Metals Corp., 658 Ohio St., Buffalo 3, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Wallingford Steel Co., Wallingford, Ct.

SHEET, ZINC

Illinois Zinc Co., 2959 W. 47th St., Chicago 32, Ill.
New Jersey Zinc Co., 160 Front St., N.Y.C. 7

SHEET METAL ASSEMBLY & MANUFACTURING

Ackermann Mfg. Co., Wheeling, W. Va.
Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
Burt Mfg. Co., 932 S. High St., Akron 11, O.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Dean Products Co., 1042 Dean St., Brooklyn 16, N.Y. (p. 152)
Erie Art Metal Co., 1602 E. 18 St., Erie, Pa. (Parts)
Evans Mfg. Co., 460 S. 10th Ave., Mt. Vernon, N.Y.
Falstrom Co., 13 Falstrom Ct., Passaic, N.J.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
Maysteel Products, Inc., 710 N. Plankinton Ave., Milwaukee 3, Wis.
Mullins Mfg. Corp., S. Ellsworth St., Salem, O.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
New York Iron Roofing & Corrugating Co., Inc., 94 1st St., Jersey City 2, N.J.
Palmer Bee Co., 1701 Poland Ave., Detroit 12, Mich.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Robinson Aviation, Inc., Teterboro, N.J.
Stanley Wks., 195 Lake St., New Britain, Ct.
Wilson Refrigeration Inc., S. Main St., Smyrna, Del.

SHELLS, DRAWN OR SEAMLESS (See SHAPES, DRAWN)

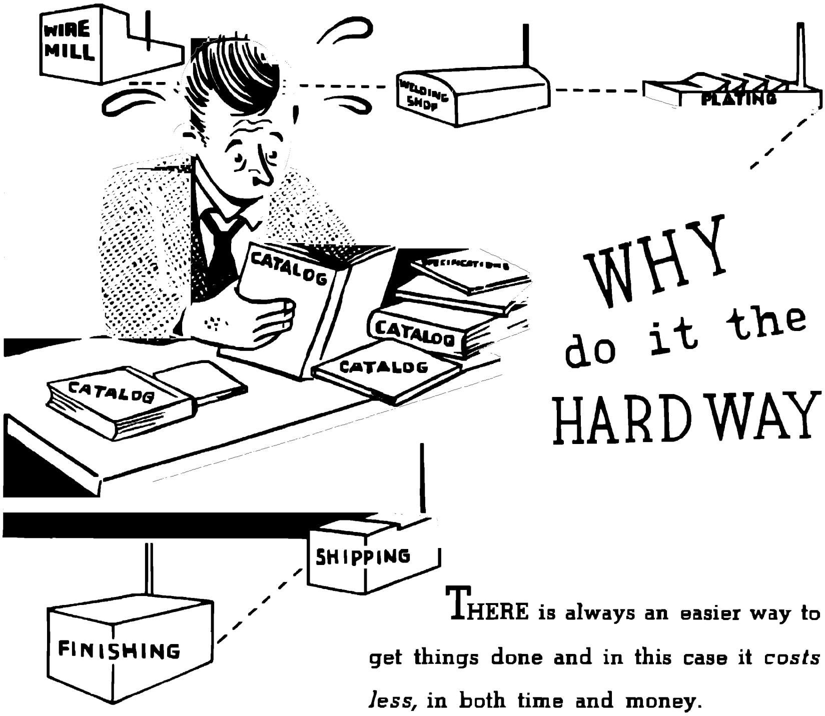
SHELVES, REFRIGERATOR (See also RACKS; also WIRE FORMING)

Bauer Bros. Co., 1774 Sheridan Ave., Springfield, O.
Collis Co., Box 231, Clinton, Ia. (p. 174)

(Continued)

THE COLLIS CO.
CLINTON, IOWA
Manufacturers of
- Refrigerator Shelving
- Refrigerator Baskets
- Freezer Baskets
- Dividers and Separators
- Wire Guards
- Wire Grills
- All Types of Wire Specialties

Write or Wire Regarding Our Complete Facilities



WHY do it the HARD WAY?

THERE is always an easier way to get things done and in this case it *costs less,* in both time and money.

As contract suppliers of fabricated wire products, welded steel wire shelves, baskets, wire shapes, we are *completely* equipped to handle any job. Starting with a design *engineered* for economical production, we produce with automatic machines, then finish in our own plant to *your* specification.

We *know* we can deliver a better product and we know we can save you money. Why not let us try?

SUB CONTRACTS ON WAR GOODS

We have 385,000 square feet of plant equipped with modern machines, devoted to welded wire fabrication. Our facilities are available to you for your sheet metal, formed wire and welded wire assembly requirements. All operations in one plant in non-strategic area, well served by all forms of major transportation. Explanatory facilities folder will be sent upon request.

UNION STEEL PRODUCTS COMPANY
Wire Products Division • ALBION, MICHIGAN

Fabricators of Welded Wire Products—Manufacturers of steel wire Pallets and PALLETAINERS for material handling and shipment—WENDWAY Conveyors for plant efficiency

SHELVES, REFRIGERATOR (Continued)

Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Reynolds Metals Co, 2000 S. 9th St, Louisville 1, Ky
Sneath Glass Co., Hartford City, Ind.
Amos Thompson Corp, Edinburgh, Ind
E. H. Titchener & Co, Binghamton, N J

Union Steel Products Co., 448 Pine St., Albion, Mich. (*p. 175*)
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
United Steel & Wire Co., Battle Creek, Mich.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

SHIELDS, EXPANSION (See ANCHORS)

SHIMS

Chase Brass & Copper Co., 236 Grand St, Waterbury 91, Ct.
Detroit Stamping Co 418 Midland Ave, Detroit 3, Mich
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Laminated Shim Co., Inc., Union St., Glenbrook, Ct
National Motor Bearing Co., Inc., Redwood City, Cal.
Van Cleef Bros, Inc, 7800 S Woodlawn Ave, Chicago 19, Ill.

SHOCK ABSORBERS (See VIBRATION ABSORBING BASES)

SHUTTERS, AUTOMATIC

Allen Ventilating Div., Production Planning Co, 704 Woodward, Rochester, Mich
American Coolair Corp, 3606 Mayflower St, Jacksonville 3, Fla
Arex Co., 333 N. Michigan Ave, Chicago 1, Ill
Barber-Colman Co Rockford, Ill
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 11, O.
Diehl Mfg. Co, 1152 Finderne Ave Somerville, N J
Electrovent Fan & Mfg Co, 812 W Lake St, Chicago 7, Ill.
General Bower Co, Ferris Ave, Morton Grove, Ill
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
H. H. Robertson Co, 2400 Farmers' Bank Bldg, Pittsburgh 22, Pa.
V. E. Sprouse Co, Inc, Columbus, Ind
L. J. Wing Mfg Co, 151 W. 14th St., N.Y.C. 11

SIGHT GLASSES (See LIQUID INDICATORS)

SIGNALS, ELECTRIC

Ashton Valve Co, 43 Kendrick St, Wrentham, Mass
Automatic Control Co, 1005 University Ave, St. Paul 4, Minn
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co, 4414 Wayne Ave., Phila. 44, Pa.
Colloid Equip Co, 50 Church St, N.Y.C.
Coral Designs, 12-16 Astoria Blvd, Long Island City 2, N.Y.
Enterprise Products Co., P.O. Box 577, Freeport, Ill.
Fischer & Porter Co, Hatboro, Pa.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. (*p. 21*)
McDonnell & Miller, Inc, 3500 N. Spaulding Ave, Chicago 18, Ill.
Magnetrol, Inc., 2110 S. Marshall Blvd., Chicago 23, Ill.
Manning, Maxwell & Moore, Inc, 11 Elias St, Bridgeport 2, Ct.
Marshalltown Mfg Co., Marshalltown, Ia.
Mason Neilan Regulator Co, 1190 Adams St., Boston 24, Mass.
Panellit, Inc., 7212 N. Clark St., Chicago 26, Ill
H. A. Phillips & Co., 3255 W. Carroll Ave. Chicago 24, Ill. (*p. 142*)
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Russell & Stoll Co., Inc., 125 Barclay St., N.Y.C.
Signal Engrg. & Mfg Co., 154 W. 14th St., N.Y.C. 11
Standard Thermometer, Inc, 952 Dorchester Ave, Boston 25, Mass.
Taylor Instrument Cos., 95 Ames St, Rochester 1, N.Y
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Ill.
Yarnall-Waring Co., Chestnut Hill, Phila 18, Pa.

SIGNS (See also NAME PLATES)

Acromark Co., 5 Morrell St., Elizabeth 5, N.J.
Anderson & Sons, Inc N. Elm St, Westfield, Mass
Artkraft Mfg. Corp., Kibby St. & D.T.&I. R.R., Lima, O
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Ingram-Richardson Mfg Co, Beaver Falls, Pa
Mine Safety Appliances Co., Braddock, Thomas & Meade Sts, Pittsburgh, Pa.
Newman Bros, Inc., 666 W 4th St., Cincinnati 3 O
Amos Thompson Corp., Edinburg, Ind.

SILICA GEL (See DEHYDRANTS)

SILVER

Baker & Co, Inc, 113 Astor St Newark 5, N J
E I. du Pont de Nemours & Co, Inc Wilmington 98 Del (Powder)
Handy & Harman, 82 Fulton St, N Y.C 7
P. R. Mallory & Co, Inc., 3029 E. Washington St, Indpls., Ind.

SLEEVES, CYLINDER

McQuay-Norris Mfg. Co, 2320 Marconi Ave, St Louis 10, Mo.

SLIDES, DRAWER

Brasco Mfg. Co., Harvey, Ill.
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Standard-Keil Hardware Mfg. Co., Inc., 630 B'way N Y.C 12
E R Wagner Mfg Co, 4001 N 32nd St, Milwaukee, Wis.

SODA FOUNTAINS, FOUNTAINETTES & EQUIPMENT (See also CARBONATORS, etc.)

Ace Cabinet Corp, 110 E. 42nd St., N.Y.C. 17
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 4 Ill.
Horace A. Carter, Inc., 16 E. Marshall St., Richmond 19, Va.
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Hudson Industries, Inc, 4400 St. Aubin Ave, Detroit 7, Mich
Scientific Research Co, 1618 N Vancouver, Portland 12 Ore
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Nelson Mfg. Co., 4016 N. Union St., St. Louis 15, Mo.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.

SODIUM CHLORIDE

Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

SODIUM DICHROMATE (See WATER TREATING MATERIALS)

SOFTENERS, WATER (See WATER TREATING MATERIALS)

SOLDER (See also SOLDER, SILVER)

All-State Welding Alloys Co., Inc., 273 Ferris Ave., White Plains, N.Y.
Division Lead Co , 836 Kinzie St., Chicago 22, Ill
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Federated Metals Div., American Smelting & Refining Co , 120 B'way, N.Y C 5
Handy & Harman, 82 Fulton St , N.Y C. 7.
Lenk Mfg Co , Newton Lower Falls 62, Mass.
Mueller Brass Co , 1925 Lapeer Ave Port Huron, Mich
National Lead Co., 111 Broadway, N.Y.C. 6
Northern Indiana Brass Co., 935 Plum St., Elkart, Ind
Northwest Lead Co., 2700-16th Ave., S W., Seattle 4, Wash.
United Wire & Supply Corp , 1497 Elmwood Ave , Providence 7, R I.

SOLDER, SILVER

Air Reduction Sales Co , 60 E 42nd St , N.Y.C 17
All-State Welding Alloys Co , Inc., 273 Ferris Ave., White Plains, N.Y.
American Platinum Wks , 231 N.J R R Ave , Newark 5, N J
Baker & Co , Inc , 113 Astor St Newark 5, N J
Chase Brass & Copper Co , 236 Grand St., Waterbury 91, Ct.
Division Lead Co , 836 Kinzie St , Chicago 22, Ill
Eutectic Welding Alloys Corp , 40 Worth St., N Y C. 13
General Plate Div , Metals & Controls Corp , Attleboro, Mass.
Handy & Harman, 82 Fulton St , N.Y.C 7
Chas W Krieg Co , 48 Dickerson St , Newark 4, N.J.
Lucas Milhaupt Engrg Co , 5051 S Lake Drive, Milwaukee, Wis
P. R. Mallory & Co , Inc , 3029 E Washington St., Indpls Ind.
Mueller Brass Co , 1925 Lapeer Ave Port Huron Mich
National Lead Co , 111 Broadway, N.Y C. 6
United Wire & Supply Corp , 1497 Elmwood Ave , Providence 7, R I.

SOLENOID VALVES, REFRIGERANT (A—Ammonia; B—Other refrigerants)

(A,B) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
(B) A P Controls Corp 2450 N 32nd St Milwaukee 10 Wis
(A,B) Automatic Switch Co , 391 Lakeside Ave , Orange, N.J.
Dean W Davis & Co , 3418 N Milwaukee Ave Chicago 41, Ill
(B) Detroit Lubricator Co , 5900 Trumbull Ave , Detroit 8, Mich.
(A,B) Frick Co., Waynesboro, Pa. *(p. [illegible]1)*
(A,B) General Controls Co., 801 Allen Ave., Glendale, Cal. *(p. 74)*
(B) Hays Mfg. Co , 12th & Liberty Sts., Erie, Pa.
(A,B) Henry Valve Co., Melrose Park, Ill. *(p 208)*
(A,B) Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 189)*
(A,B) James P. Marsh Corp Skokie, Ill
(B) Penn Elec. Switch Co., Goshen, Ind. *(p 69)*
(A,B) Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sarco Co., Inc , 350-5th Ave., N.Y.C. 1 *(p. 167)*
(A,B) Sporlan Valve Co , 7525 Sussex Ave., St Louis 17, Mo

SOLENOID VALVES, WATER, BRINE, etc.

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
A-P Controls Corp , 2450 N 32nd St , Milwaukee 10 Wis.
Automatic Switch Co., 391 Lakeside Ave., Orange, N.J.
Barber-Colman Co , Rockford, Ill
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Davis Regulator Co , 2541 S. Washtenaw Ave Chicago 8, Ill.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
General Electric Co., 1 River Rd , Schenectady 5, N.Y.
Erickson Specialty Co., 10 Cayuga St., Cohoes, N.Y.
Hays Mfg Co., 21st & Liberty Sts , Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p 208)*
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p 189)*
James P Marsh Corp , Skokie, Ill
W H. Nicholson & Co , 209 Oregon St., Wilkes-Barre Pa
Penn Elec. Switch Co., Goshen, Ind. *(p 69)*
Fred D. Pfening Co 1075 W 5th Ave , Columbus 8, O
Sarco Co., Inc., 350-5th Ave , N.Y.C. 1 *(p. 167)*
Sporlan Valve Co , 7525 Sussex Ave , St Louis 17, Mo
Supreme Elec. Products Co., 194 Vassar St , Rochester 7, N.Y.

SPEED CHANGERS (See also DRIVES, VARIABLE SPEED)

Allis-Chalmers Mfg Co , Milwaukee 1, Wis
Boston Gear Works, Quincy 71, Mass
Chicago Die Casting Mfg Co , 2500 W Monroe St , Chicago 12 Ill
DeLaval Steam Turbine Co , 853 Nottingham Way, Trenton 2, N.J.
Electric Machinery Mfg. Co , 1338 Tyler St N E Minneapolis 13, Minn
Fafnir Bearing Co , 37 Booth St., New Britain, Ct
General Elec. Co 1 River Rd , Schenectady 5, N.Y.
Janette Mfg Co 556 W Monroe St , Chicago 6, Ill
W. A Jones Foundry & Machine Co., 4401 Roosevelt Rd , Chicago 24, Ill.
Lima Elec Motor Co , Findlay Rd , Lima O
Link-Belt Co , 2045 W Hunting Park, Phila 40, Pa
Metron Instrument Co , 2541 So. Washtenaw Ave Chicago 8, Ill.
Ohio Gear Co 1333 E 179 St Cleveland, O
Palmer Bee Co , 1701 Poland Ave Detroit 12, Mich
Philadelphia Gear Wks , Inc , G St & Erie Ave , Phila 20, Pa
Reliance Elec & Engrg Co , 1088 Ivanhoe Rd., Cleveland 10, O
Speed Control Corp , Wickliffe O
Star Elec. Motors Co , 200 Bloomfield Ave , Bloomfield, N.J.
Stephens-Adamson Mfg Co , Ridgeway Ave , Aurora, Ill
O A Sutton Corp , 1812 W. 2nd St , Wichita 1, Kan
Twin Disc Clutch Co , Racine, Wis.
Westinghouse Elec. Corp , 200 McCandless Ave., Pittsburgh 1, Pa.

SPEED NUTS & CLIPS (See FASTENERS)

SPONGE RUBBER PARTS (See RUBBER PRODUCTS, CELLULAR)

SPRAY ELIMINATORS

Air & Refrigeration Corp , 475 5th Ave , New York 17, N Y.
Bahnson Co , 1001 S Marshall St , Winston-Salem, N C
Farr Co., 2615 Southwest Dr , Los Angeles 43, Cal.
Gay Engrg Co., 2730 E 11th St , Los Angeles 23, Cal
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p 74)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p 89)*
Trane Co., La Crosse, Wis. *(p 118)*

SPRAY HUMIDIFIERS (See HUMIDIFIERS)

SPRAY NOZZLES (See NOZZLES)

SPRAY PONDS (See COOLING PONDS)

SPRAY TYPE EVAPORATORS (See UNIT COOLERS)

SPRINGS

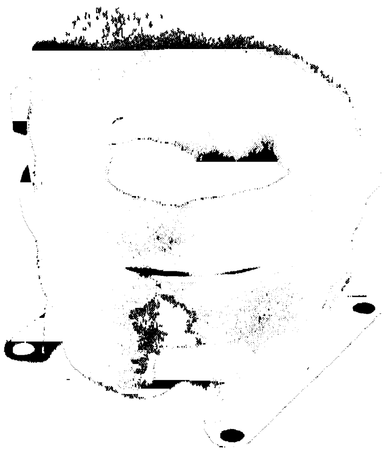
American Spring & Wire Specialty Co., 816 N. Spaulding Ave., Chicago 51, Ill.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Hunter Pressed Steel Co., 801 Maple St., Lansdale, Pa.
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

SPROCKETS

Charles Bond Co , 617-23 Arch St , Phila , Pa.
Boston Gear Works, Quincy 71, Mass
Chain-Belt Co , 1600 W Bruce St Milwaukee 4 Wis
Cyclone Fence Div , American Steel & Wire Co , U. S. Steel Corp Subsidiary, P O. Box 260, Waukegan 1, Ill.
Diamond Chain Co , Inc., 402 Kentucky Ave , Indpls 7, Ind.
Jeffrey Mfg Co , 887 N 4th St , Columbus 16 O
Link-Belt Co., 519 Holmes Ave., Indpls 6, Ind
Morse Chain Co , Div of Borg-Warner Corp , Ithaca, N Y
Ohio Gear Co 1333 E 179 St Cleveland O

STAMPINGS (See also SHAPES)

Ackermann Mfg Co , Wheeling, W Va
Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O *(p 178)*
Acme Equip Co , 205 E Broadway, Muskogee, Okla
Alloy Products Corp , 1015 Perkins Ave , Waukesha, Wis
Aluminum Goods Mfg Co , Manitowoc Wis
American Brass Co , Waterbury 88, Ct. *(p 189)*
American Central Mfg Corp , Div of Avco Corp , 18th & Columbia Sts , Connersville, Ind
American Emblem Co , Inc , P O Box 116 H Utica 1, N Y
Atlantic Steel Co , P O Box 1714 Atlanta 1, Ga
Behringer Metal Wks , Inc , 108 Jabez St , Newark 5, N J
Bettinger Enamel Corp , Metal Fabricating Div , Waltham Mass
Bingham Herbrand Co , 1062 S Post St , Toledo 6, O
Bossert Co P O Drawer 358, Utica 1, N Y
A S Campbell & Co , Inc , E Boston 28, Mass
Chase Brass & Copper Co , 236 Grand St , Waterbury 91, Ct.
Columbia Metal Stamping Co , 1900 Harvard Ave., Cleveland, O.
Commercial Shearing & Stamping Co., 1775 Logan St , Youngstown, O.
Crandal-Stone Div , Brewer-Titchener Corp , 336 Court St , Binghamton, N Y.
Dahlstrom Metallic Door Co , 435 Buffalo St., Jamestown, N.Y.
Dean Products Co., 1042 Dean St., Brooklyn 16, N.Y. *(p 152)*
Detroit Stamping Co 118 Midland Ave , Detroit 3, Mich
Erie Art Metal Co , 1602 E 18 St , Erie, Pa.
Fox Co , Fox Lane, Cin'ti 23, O
Gillian Mfg. Co , 7750 Dubois Ave , Detroit 11, Mich
Globe Stamping Div , Hupp Corp , 1250 W. 76th St , Cleveland 2, O
L F Grammes & Sons, Inc , 365 Union St., Allentown, Pa.
Greene Mfg. Co , 1028 Douglas Ave , Racine, Wis
Greene Mfg Co , Inc , 1028 Douglas Ave Racine, Wis
Heintz Mfg Co , Front St & Olney Ave , Phila 20, Pa
Hoosier Cardinal Corp , 601 W Lichel Ave., Evansville 7 Ind.
Houdaille-Hershey Corp , 1900 Foss Park Ave , North Chicago, Ill. *(p 162)*
Hugo Mfg Co , 49 Ave W. & Superior St , Duluth 7, Minn
Hunter Pressed Steel Co , 801 Maple St , Lansdale, Pa
Industrial Wire Cloth Products Corp , Wayne, Mich *(p 88)*
Ingram Richardson Inc Frankfort Ind
Lansing Stamping Co , 1159 S Pennsylvania Ave , Lansing 2, Mich.



Specify
Acklin High Quality
Stampings for your Product.
Pressed Metal Parts and Assemblies Built to Your Specifications
- Stamped Hermetic Housings
- Compressor Bases
- Liquid Receivers
- Stamped Panels

200 Presses Ranging From 10 Tons to 1000 Tons
Complete Die Shop
Controlled Atmosphere Brazing
Our 130,000 Sq. Ft. Modern Plant is Geographically Located to Serve you by Rail or Truck
Send us Your Prints and Specifications
1911—FORTY YEARS OF SERVICE TO INDUSTRY—1951
The Acklin Stamping Company
Factory & Office: 1929 Nebraska Ave., Toledo 7, Ohio
Representatives: New York—Chicago—Cleveland—Indianapolis—Milwaukee
Buffalo—Lansing—Detroit—Cincinnati

Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Melrath Supply & Gasket Co., Inc., Tioga & Memphis Sts., Phila. 34, Pa.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
Motors Metal Mfg. Co., 5936 Milford Ave., Detroit 10, Mich.
Mullins Mfg. Corp., S. Ellsworth St., Salem, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill.
Northern Engraving & Mfg. Co., 4th & Vine Sts., La Crosse, Wis.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Powell Pressed Steel Co., Hubbard, O.
Quality Hardware & Machine Div., 5847 North Ravenswood, Chicago 26, Ill.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
Robinson Aviation, Inc., Teterboro N.J.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
V. E. Sprouse Co., Inc., Columbus, Ind.
Stanley Wks., 195 Lake St., New Britain, Ct.
Superior Spinning & Stamping Co., 4057 Fitch Rd., Toledo 12, O.
Swift Mfg. Co., Inc., 1455 E. Nine Mile Rd., Hazel Park, Mich.
E. H. Titchener & Co., 76 Clinton St., Binghamton, N.Y.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O.
Transue & Williams Steel Forging Corp., Alliance, O.
E. R. Wagner Mfg. Co., 4001 N. 32nd St., Milwaukee, Wis.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
Wheeling Steel Corp., Wheeling, W. Va.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16. Mich.
Worcester Pressed Steel Co., 100 Barber Ave., Worcester 6, Mass.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

STANDARDS. SHELF (See SUPPORTS)

STAPLES

Fastener Corp., 860 Fletcher St., Chicago 14, Ill.
Heller Co., 2135 Superior Ave., N.E., Cleveland 4, O.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
E. H. Titchener & Co., 76 Clinton St., Binghamton, N.Y.

STARTERS, MOTOR

Allen-Bradley Co., Milwaukee 4, Wis.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St. Hartford 6, Ct. (Manual)
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Electric Controller & Mfg. Co., 2700 E. 79th St., Cleveland 4, O.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
Fitch, Allen & Co., 6314 N.W. Highway, Chicago 31, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Russell & Stoll Co., Inc., 125 Barclay St., N.Y.C.
Sarco Co., Inc., 350 5th Ave., N.Y.C. 1 *(p. 167)*
Spencer Thermostat Div., Metals & Controls Corp., 34 Forest St., Attleboro, Mass.
Square D Co., Inc., 6060 Rivard St., Detroit 11, Mich.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Westinghouse Elec. Corp., Beaver, Pa.
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.

STEAM COILS (See AIR CONDITIONING COILS)

STEAM JET VACUUM COOLING SYSTEMS (See also EJECTORS)

Croll-Reynolds Engrg. Co., Inc., 17 John St., N.Y.C.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Ingersoll Rand Co., 11 Broadway, N.Y.C. 4
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Schutte & Koerting Co., Cornwells Heights, Bucks County, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*

STEAM SEPARATORS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Climax Controls Div., Black, Sivalls & Bryson, Inc., Box 1529, Tulsa 1, Okla.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Johnson Corp., 805 Wood St., Three Rivers, Mich.
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
W. H. Nicholson & Co., 203 Oregon St., Wilkes-Barre, Pa.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Wilson Engrg. Corp., 122 S. Michigan Ave., Chicago 3, Ill.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

STEAM TRAPS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Armstrong Machine Wks., 831 Maple St., Three Rivers, Mich. *(p. 160)*
V. D. Anderson Co., 1935 W. 96th St., Cleveland 2, O.
Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain 30, Mass.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Climax Controls Div., Black Sivalls & Bryson, Inc., Box 1529, Tulsa 1, Okla.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
C. A. Dunham Co., 400 W. Madison St., Chicago 6, Ill.
Hoffman Specialty Co., Inc., 1001 York St., Indpls., Ind.
Marsh Heating Equip. Co., 3501 Howard St., Skokie, Ill.
W. H. Nicholson & Co., 203 Oregon St., Wilkes-Barre, Pa.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Trane Co., La Crosse, Wis. *(p. 118)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
Warren Webster & Co., Camden, N.J.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 12, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

STEAM TURBINES

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Dean Hill Pump Co., 4000 E. 16th St., Indpls. 7, Ind.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Elliott Co., Jeannette, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Terry Steam Turbine Co., Hartford 1, Ct.
Westinghouse Elec. Corp., S. Phila. 1, Pa.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*

STEEL (See particular mill forms, i.e., BAR, SHEET, etc.)

STEEL, COMPRESSOR VALVE

Athenia Steel Div., National-Standard Co., Clifton, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
Sandvik Steel, Inc., 111-8th Ave., N.Y.C. 11 (Swedish Steel)

STEEL PLATE FABRICATION

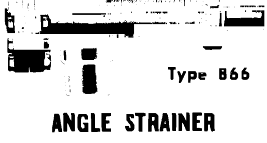
Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Central Iron & Steel Co., Harrisburg, Pa.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Falstrom Co., 13 Falstrom Ct., Passaic, N.J.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(P. 39)*
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 158)*
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.J.
McNamara & Co., Inc., 2700 Manokin St., Baltimore 30, Md.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

STEEL, STRUCTURAL (See SHAPES, STRUCTURAL)

STERILIZERS, AIR (See AIR PURIFICATION EQUIPMENT)

HENRY

STRAINERS


Type 866

ANGLE STRAINER

Has reinforced monel screen, brass shell, forged brass end caps with integral fittings. Silver brazed joints. 8 bolt distortion-proof access flange. Screen area 23 to 105 sq. in. Sizes ⅜″ thru 2⅝″ O.D. solder.

WRITE FOR CATALOG

HENRY VALVE CO.
Melrose Park, Illinois, Suburb of Chicago

APPROVED FOR USE BY THE ARMY • NAVY • COAST GUARD AND THE MARITIME COMMISSION.

STILLS, AMMONIA

Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh, Pa.

STRAINERS, OIL

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
American District Steam Co., Bryant St., N., Tonawanda, N.Y.
American Flexible Coupling Co., 1808 Pittsburgh Ave., Erie, Pa.
V. D. Anderson Co., 1935 W. 96th St., Cleveland 2, O.
Anthony Co., 47-33-5th St., Long Island City, N.Y.
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. *(p. 209)*
Bethlehem Steel Co., Bethlehem, Pa.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Elliott Co., Jeannette, Pa.
Fischer & Porter Co., Hatboro, Pa.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Industrial Wire Cloth Products Corp., Wayne, Mich. *(p. 88)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Schutte & Koerting Co., 12 Cornwells Heights, Bucks County, Pa.
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

STRAINERS, REFRIGERANT

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
American Flexible Coupling Co., 1808 Pittsburgh Ave., Erie, Pa.
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 149)*
V. D. Anderson Co., 1935 W. 96th St., Cleveland 2, O.
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Cee-Kleer Products, Inc., 947 W. 6th St., Cincinnati 3, O.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 64)*
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 180)*
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 183)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Industrial Wire Cloth Products Corp., Wayne, Mich. *(p. 88)*
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.

Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 2 5)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 86)*
Madden Brass Products Co., 1111 N. Franklin St., Chicago 10, Ill.
James P. Marsh Corp., Skokie, Ill.
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Mueller Brass Co., Port Huron, Mich.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 142)*
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Remco, Inc., Zelienople, Pa. *(p. 87)*
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Wabash Mfg. Co., 2300 South Western Ave., Chicago 8, Ill.
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill

STRAINERS, STEAM, AIR, WATER, etc.

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
American District Steam Co., Bryant St., N. Tonnawanda, N.Y.
American Flexible Coupling Co., 1808 Pittsburgh Ave., Erie, Pa.
A. A. Anderson Co., 1935 W. 96th St., Cleveland 2, O.
Armstrong Machine Works, 831 Maple St., Three Rivers, Mich. *(p. 160)*
Bethlehem Steel Co., Bethlehem, Pa
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Elliott Co., Jeannette, Pa.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Fischer & Porter Co., Hatboro, Pa.
Foster Engrg. Co., 835 Lehigh Ave., Union, N.J
General Controls Co., 801 Allen Ave., Glendale, Cal. *(p. 64)*
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 189)*
Hoffman Specialty Co., Inc., 1001 York St., Indianapolis, Ind
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 183)*
Industrial Wire Cloth Products Corp., Wayne, Mich. *(p. 88)*
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill
James P. Marsh Corp., Skokie, Ill.
Jos. A. Martorello & Co., 229 N. 14th St., Phila. 7, Pa
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Monarch Mfg. Wks., Inc., Salmon & Westmorland Sts., Phila. 34, Pa
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Brass Co., Port Huron, Mich.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
Neptune Meter Co., 50. W. 50th St., N.Y.C. 20
W. H. Nicholson & Co., 209 Oregon St., Wilkes-Barre, Pa.
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Schutte & Koerting Co., Cornwells Heights, Bucks County, Pa.
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Trane Co., La Crosse, Wis. *(p. 118)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa
J. A. Zurn Mfg. Co., Erie, Pa.

STRAINERS, WELL

A. D. Cook, Inc., Lawrenceburg, Ind.
Capital Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.

STRAPS, TUBE (See CLAMPS)

STRIP, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Fairmont Aluminum Co., Fairmont, W. Va.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.

STRIP, BERYLLIUM COPPER

Athenia Steel Div., National Standard Co., Clifton, N.J
Beryllium Corp., P.O. Box 1462, Reading, Pa
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

STRIP, BRASS, BRONZE, COPPER

American Brass Co., Waterbury 88, Ct. *(p. 189)*
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91 Ct.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburgh 19, Pa
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind. (Alloy)
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Western Brass Mills, Div. of Olin Industries, Inc., E. Alton, Ill.

STRIP, INCONEL, MONEL, NICKEL

Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
International Nickel Co., 67 Wall St., N.Y.C. 5

STRIP, STEEL

Acme Steel Co., 2840 Archer Ave., Chicago 8, Ill.
Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Armco Steel Corp., Middletown, O.
Athenia Steel Div., National-Standard Co., Clifton, N.J.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Sandvik Steel Inc., 111-8th Ave., N.Y.C. 11
Stanley Works, 195 Lake St., New Britain, Ct.

(Continued)

STRIP, STEEL (Continued)

Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Wallingford Steel Co., Wallineford, Ct.
Weirton Steel Co., Weirton W. Va.
Wheeling Steel Corp., Wheeling, W. Va.

STRIP, STEEL, COATED, CLAD or GALVANIZED

Acme Steel Co., 2840 Archer Ave., Chicago 8, Ill.
Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Armco Steel Corp., Middletown, O.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Steel Corp., Carnegie, Pa.
Weirton Steel Co., Weirton, W. Va.
Wheeling Steel Corp., Wheeling, W. Va.

STRIP, STEEL, STAINLESS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Armco Steel Corp., Middletown, O.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Rigidized Metals Corp., 658 Ohio St., Buffalo 3, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Sandvik Steel, Inc., 111-8th Ave., N.Y.C. 11
Superior Steel Corp., Carnegie, Pa.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm. Penn Place Bldg., Pittsburgh 30, Pa.
Wallingford Steel Co., Wallingford, Ct.

STRIP, ZINC

Illinois Zinc Co., 2959 W. 47th St., Chicago 32, Ill.
New Jersey Zinc Co., 160 Front St., N.Y.C. 7

STRIP HEATERS (See HEATER ELEMENTS)

STRUCTURAL SHAPES (See SHAPES, STRUCTURAL)

STRUCTURAL STEEL FABRICATION

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Black, Sivalls & Bryson, Kansas City 3, Mo.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.Y.
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
McNamara & Co., Inc., 2700 Manokin St., Baltimore 30, Md.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.

STRUCTURE, PREFABRICATED

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Lindsay Structure, Inc., 5000 W. Dempster St., Skokie, Ill.

STUDS (See BOLTS)

SUCTION PRESSURE REGULATING VALVES (See also TWO TEMPERATURE VALVES)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 149)*
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Climax Controls Div., Black, Sivalls & Bryson, Inc., Box 1529, Tulsa 1, Okla.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Hays Corp., Michigan City, Ind.
Hubbell Corp., P.O. Box 700, Hawley Rd., Mundelein, Ill. *(p. 183)*
James P. Marsh Corp., Skokie, Ill.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p. 89)*
York Corp., York, Pa. *(p. 52)*

SULFUR DIOXIDE

Ansul Chemical Co., Marinette, Wis. *(p. 115)*
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal.
Virginia Smelting Co., W. Norfolk, Va. *(p. 35)*
Western Div., Dow Chemical Co., 350 Sansome St., San Francisco 4, Cal.

SUPPORTS, SHELF (See HARDWARE SPECIALTIES)

SUPPORTS, TUBE (See CLAMPS)

SURGE TANKS (See ACCUMULATORS)

SWING JOINTS

Barco Mfg. Co., 1801 Winnemac Ave., Chicago 40, Ill.
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Chicago Screw Co., 2701 Washington Blvd., Bellwood, Ill.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Flexo Supply Co., Inc., 4649 Page Blvd., St. Louis 13, Mo.
Mayson Mfg. Co., 4332 Horatio St., Detroit 10, Mich.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Water Cooling Corp., 71 Nassau St., N.Y.C. 7

SWITCHES, ELECTRIC (See also FLOAT SWITCHES; also STARTERS, MOTOR; also TIME SWITCHES)

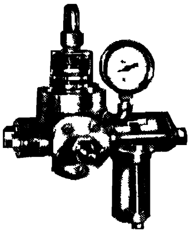
Allen Bradley Co., Milwaukee 4, Wis.
American Relay & Controls, Inc., 4901 Flournoy St., Chicago 44, Ill.
Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St. Hartford 6, Ct.
Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Automatic Switch Co., 391 Lakeside Ave., Orange, N.J.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Bryant Elec. Co., Div. of Westinghouse Elec. Corp., 1421 State St., Bridgeport 2, Ct.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Clarostat Mfg. Co., Inc., 130 Clinton St., Brooklyn 2, N.Y
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Cornish Wire Co., Inc., Rutland, Vt.

(Continued)

Hubbell Corporation

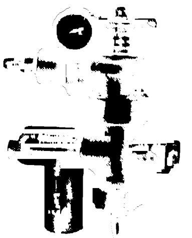
P.O. Box 700, Hawley Rd.
Mundelein, Ill.

Designers and Manufacturers of Automatic Control Valves For All Refrigerants



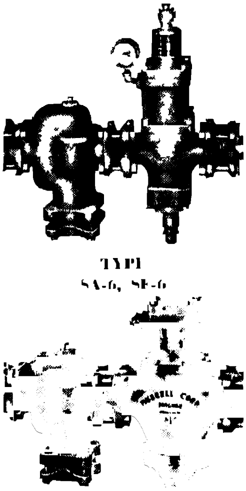
TYPE
SA-5, SF-5

Type SA 5, SA 6, SF 5 and SF 6 Back Pressure Regulating Valves are of the conventional type used to maintain a constant evaporator pressure.



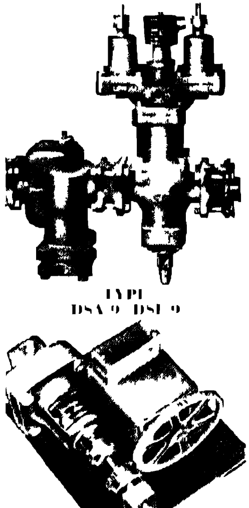
TYPE
SA 7, SF 7

Type SA 7, SA 8, SF 7 and SF 8 Combination Back Pressure Regulator and Stop Valves. This regulator is of the conventional type used to maintain a constant evaporator pressure and the addition of a small electric pilot valve built into the head makes it a suction stop valve.



TYPE
SA-6, SF-6

The DSA 9 and DSF 9 is a dual regulator which will control evaporators with two load conditions requiring different refrigerant temperatures. The diaphragms in the dual head may be set for any two evaporator pressures and will automatically change from one to the other by the opening or closing of the electric pilot valve which is built into the head.

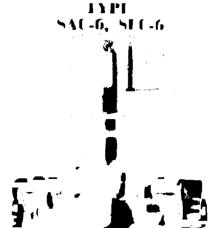


TYPE
DSA 9, DSF 9

The SAC 6 and SFC 6 valves are of the compensating type and are used where a constant temperature is desired in the medium being cooled. These valves will increase or decrease the evaporator pressure to compensate for the increase or decrease of the cooling load. These valves are made to operate with air or electricity.

Type "T"

The Type "T" suction stop valve is used on installations where automatic suction line control is required. It is operated by high pressure gas and its construction makes a tight closing valve and its dependability far surpasses the conventional magnetic stop valve.



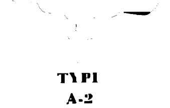
TYPE
SAC-6, SFC-6

All of the valves have built in opening stem which eliminates the by passes usually used for manual operation. Available with either screwed, welding type flanges or copper tube connections in sizes from [illegible]" to 8" inclusive.

Solenoid Valves from ⅛" to [illegible] inclusive for liquids and gases with composition seat discs, readily renewable coils for any electrical characteristics, built in lifting stem standard. Available with either screwed, welding flanges or copper tube connections.

Strainers are available in all sizes for liquid and gas with very large screen areas and arranged to bolt directly to valve or with screwed, welding flange or copper tube connections for installation wherever strainer is necessary.

Write for complete information on these and our many additional Refrigeration Controls and Accessories.



TYPE
A-2

TYPE
A 3, F-4

SWITCHES, ELECTRIC (Continued)

Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Dura Kool, Inc., 1010 N. Main St., Elkhart, Ind.
Fitch, Allen & Co., 6314 N.W. Highway, Chicago 31, Ill.
General Control Co., 1200 Soldiers Field Rd., Boston 34, Mass.
General Elec. Co., 1285 Boston Ave., Bridgport 2, Ct.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
McGill Mfg. Co., Inc., Valparaiso, Ind.
Magnetrol, Inc., 2110 S. Marshall Blvd., Chicago 23, Ill.
P. R. Mallory & Co.,Inc., 3029 E. Washington St., Indpls., Ind.
W. L. Maxson Corp., 460 W. 34th St., N.Y.C. 1
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Micro Switch Div., First Industrial Corp , Freeport, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p* 67)
Nu-Switch Div., Chase-Shawmut Co., 38 Pequit St., Canton, Mass.
Ohmite Mfg. Co., 4835 W. Fluornoy St., Chicago 44, Ill.
Penn Elec. Switch Co., Goshen, Ind. (*p*. 69)
Pent Elec. Co., Inc., 634 Michigan Trust Bldg., Grand Rapids 2, Mich.
Photoswitch, Inc., 77 B'way, Cambridge 42, Mass
Ranco, Inc., 601 W. 5th Ave., Columbus 1, O. (*p*. 68)
Russell & Stoll Co., Inc., 125 Barclay St., N.Y.C.
Soreng Mfg. Corp , 9555 Eden Ave., Schiller Park, Ill.
Spencer Thermostat Div., Metals & Controls Corp., 34 Forest St., Attleboro, Mass.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Unimax Switch Div. of the W. L. Maxson Corp., 460 W. 34th St., N.Y.C. 1
United-Carr Fastener Corp., 31 Ames St., Cambridge 42, Mass.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Westinghouse Elec. Corp., Beaver, Pa.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5. N.Y.

SWITCHGEAR

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Automatic Switch Co., 391 Lakeside Ave., Orange, N.J.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
I-T-E Circuit Breaker Co., 19th & Hamilton Sts., Phila. 30, Pa.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

SWIVEL JOINTS (See SWING JOINTS)

SYNTHETIC RUBBER (See RUBBER PRODUCTS)

TACHOMETERS

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct. (Recording, Elec.)
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Corbin Screw Corp., New Britain, Ct.
Eclipse-Pioneer Div., Bendix Aviation Corp., Teterboro, N.J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Ideal Industries, Inc., Sycamore, Ill.
Jones Motrola Corp., 432 Fairfield Ave., Stamford, Ct.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Metron Instrument Co., 432 Lincoln St., Denver 9, Col.
Reliance Elec & Engrg. Co, 1088 Ivanhoe Rd., Cleveland 10, O.
Veeder-Root, Inc., Garden & Sargeant Sts., Hartford 2, Ct. (Laboratory only)
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

TAGS (See NAME PLATES)

TANKS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O.
American District Steam Co., Bryant St., N., Tonawanda, N.Y.
Arrow Tank Co., Inc., 16 Barnett St., Buffalo 15, N.Y. (Wood)
Atlas Mineral Products Co., Mertztown, Pa.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Baker Refrigeration Corp., S. Windham, Me. (*p*. 63)
Berlin Chapman Co., Berlin, Wis.
Bethlehem Steel Co., Bethlehem, Pa.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Black, Sivalls & Bryson, Kansas City 3, Mo.
Bossert Co., P.O. Drawer 358, Utica 1, N.Y.
Central Iron & Steel Co., Harrisburg, Pa.
Cleveland Coppersmithing Wks., 5500 Stone Ave., Cleveland 2, O.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p*. 49)
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. (*p*. 31)
Esco Cabinet Co., West Chester, Pa.
Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. (*p*. 90)
Frick Co., Waynesboro, Pa. (*p*. 51)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p*. 39)
Joseph Kopperman & Sons, 312 New St., Phila. 6, Pa.
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.J.
Lehigh Fan & Blower Co., 128 Linden St., Allentown, Pa.
McNamara & Co., Inc., 2700 Manokin St., Baltimore 30, Md.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Nukem Products Corp., 111 Colgate Ave., Buffalo 20, N.Y.
O'Brien Industries, 84 Bishop St , Jersey City 4, N.J.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p*. 58)
Pittsburgh-Des Moines Construction Co., Neville Island, Pittsburgh 25, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St. Etna P.O., Pittsburrh, 23, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Edw. W. Renneburg & Sons Co., 2639 Boston Ave., Baltimore 24, Md.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Victor Products Corp., Hagerstown, Md.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p*. 33)
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wildman Boiler & Tank Co., 3026 Carroll Ave., Chicago 12, Ill.
York Corp., York, Pa. (*p*. 52)

TANKS, HOT WATER

American District Steam Co., Bryant St., N., Tonawanda, N.Y.
Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Biggs Boiler Wks. Co., 1000 Bank, Akron 5, O.
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City, 7, N.J.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p*. 58)

Rheem Mfg. Co., 570 Lexington Ave., N.Y.C. 22
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.

TANKS, MILK HOLDING

Black, Sivalls & Bryson, Kansas City 3, Mo.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
The Heil Co., Milwaukee 1, Wis.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.

TANKS, PRESSURE (See PRESSURE VESSELS)

TANK HEADS

Ackermann Mfg. Co., Wheeling, W. Va.
Bethlehem Steel Co., Bethlehem, Pa.
Central Iron & Steel Co., Harrisburg, Pa.
Commercial Shearing & Stamping Co., 1775 Logan St., Youngstown, O.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.
Frick Co., Waynesboro, Pa. *(p. 51)*
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City, 7, N.J.
Lukens Steel Co., Coatesville, Pa.
Maysteel Products, Inc., 740 N. Plankinton Ave., Milwaukee 3, Wis.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.

TAPE, FRICTION (See FRICTION TAPE)

TAPE, PLASTIC

Minnesota Mining & Mfg. Co., 900 Fauquier St., St. Paul 6, Minn.

TAPE, RUBBER

Anchor Packing Co., 401 N. Broad St., Philadelphia 8, Pa.
Boston Woven Hose & Rubber Co., P.O. Box 1071, Boston 3, Mass.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
L. H. Gilmer Co., Div. of U. S. Rubber Co., 7230 Keystone St., Tacony, Phila. 35, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

TEES, (See FITTINGS, PIPE)

TERMINALS & TERMINAL BLOCKS

Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St. Hartford 6, Ct.
Buchanan Elec. Prod. Corp., 225 Rt. 29, Hillside, N.J.
Burke Elec. Co., Erie, Pa.
Cornish Wire Co., Inc., Rutland, Vt.
General Control Co., 1200 Soldiers Field Rd., Boston 34, Mass.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Soreng Mfg. Corp., 9555 Eden Ave., Schiller Park, Ill.
Amos Thompson Corp., Edinburgh, Ind.
Thompson-Bremer & Co., 1640 W. Hubbard St., Chicago 22, Ill.
United Mfg. & Service Co., 405 S. 6th, Milwaukee, Wis.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

TERMINALS, HERMETIC

Fusite Corp., 6028 Fernview Ave., Cincinnati 13, O.

TEST CABINETS (See REFRIGERATORS, LOW TEMPERATURE)

THERMOMETERS, ALARM

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10 O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave , Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Electric Auto-Lite Co , Instrument & Gauge Div., Toledo 1, O.
Enterprise Products Co., P.O. Box 577, Freeport, Ill.
Fischer & Porter Co., Hatboro, Pa.
H-B Instrument Co., 2646 Trenton Ave., Philadelphia 2, Pa.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Ohio Thermometer Co., 33 Walnut St., Springfield 99, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Standard Thermometer, Inc., 952 Dorchester Ave., Boston 25, Mass.
Tagliabue Instrument Corp., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, CONTROLLING (See REGULATORS, TEMPERATURE)

THERMOMETERS, INDICATING

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2, Mich.
Anderson & Sons, Inc., N. Elm St., Westfield, Mass.
Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Electric Auto-Lite Co., Instrument & Gauge Div., Toledo 1, O.
Fee & Stemwedel, Inc., 2210 Wabansia Ave., Chicago 47, Ill.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Fulton Sylphon Division, Robertshaw-Fulton Controls Co., Knoxville 4, Tenn.
Claude S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
H-B Instrument Co., 2646 Trenton Ave., Philadelphia 2, Pa.
Hays Corp., Michigan City, Ind.
Instrument & Gauge Div., Electric Auto-Lite Co., Champlain St., Toledo 1, O.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
James P. Marsh Corp., Skokie, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*

(Continued)

THERMOMETERS, INDICATING (Continued)

Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Ohio Thermometer Co., 33 Walnut St., Springfield 99, O.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti 12, O.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Rochester Mfg. Co., Inc., 100 Rockwood St., Rochester 10, N.Y.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Standard Thermometer, Inc., 952 Dorchester Ave., Boston 25, Mass.
Tagliabue Instrument Div., Weston, Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C 12
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

THERMOMETERS, HOUSEHOLD REFRIGERATOR

Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Fee & Stemwedel, Inc., 2210 Wabansia Ave., Chicago 47, Ill.
H-B Instrument Co., 2646 Trenton Ave., Philadelphia 25, Pa.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Ohio Thermometer Co., 33 Walnut St., Springfield, O.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cincinnati 12, O.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, RECORDING

Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Electric Auto-Lite Co., Instrument & Gauge Div., Toledo 1, O.
Charles Engelhard, Inc., 850 Passaic Ave., E. Newark, N.J.
Ess Instrument Co., 96 S. Washington Ave., Bergenfield, N.J.
Fischer & Porter Co., Hatboro, Pa.
Foxboro Co., 38 Neponset Ave., Foxboro, Mass.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Claud S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
Marsh Instrument Co., 3501 Howard St., Skokie, Ill.
Mason-Neilan Regulator Co., 1190 Adams St., Boston 24, Mass.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Moore Products Co., H. & Lycoming Sts., Phila. 24, Pa.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Ohio Thermometer Co., 33 Walnut St., Springfield 99, O.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Standard Thermometer, Inc., 952 Dorchester Ave., Boston 25, Mass.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Thwing-Albert Instrument Co., Penn St. & Pulaski Ave., Phila. 44, Pa.
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C 12

THERMOREGULATORS

American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
H-B Controls Co., 2646 Trenton Ave., Philadelphia 25, Pa.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.

THERMOSTATIC BI-METALS

Baker & Co., Inc., 113 Astor St., Newark 5, N.J.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
W. M. Chace Co., 1601 Beard Ave., Detroit 9, Mich.
General Plate Div., Metals & Controls Corp., Attleboro, Mass.

THERMOSTATIC EXPANSION VALVES (See EXPANSION VALVES)

THERMOSTATS (See also CONTROLS, etc.)

Allen-Bradley Co., Milwaukee 4, Wis.
Barber-Colman Co., Rockford, Ill.
Bridgeport Thermostat Div. of Robertshaw-Fulton Controls Co., 1225 Connecticut Ave., Bridgeport 1, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Fulton Sylphon Div., Robertshaw-Fulton Controls Co., Knoxville 4, Tenn.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 64)*
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Johnson Service Co., 507 E. Michigan St., Milwaukee 2, Wis.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Penn Elec. Switch Co., Goshen, Ind. *(p. 69)*
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa.
Ranco, Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 68)*
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 167)*
Spencer Thermostat Div., Metals & Controls Corp., 34 Forest St., Attleboro, Mass.

Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
United Elec. Controls Co., 71 A St., S. Boston 27, Mass.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.
Westinghouse Elec. Corp., Meadville, Pa.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

THERMOSTATS, HOUSEHOLD REFRIGERATOR

Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Ranco, Inc., 601 W. 5th Ave., Columbus 1, O. (*p. 68*)

THREADED SPECIALTIES (See SCREW MACHINE PARTS)

TIME SWITCHES (See also CONTROLS, PROGRAM, etc.)

April Showers Co., Inc., 4126-8th St., N.W., Washington 11, D.C.
Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.
Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Automatic Elec. Mfg. Co., 10 State St., Mankato, Minn.
Automatic Temperature Control Co., Inc., 5212 Pulaski Ave., Phila. 44, Pa.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Colloid Equip. Co., Inc., 50 Church St., N.Y.C. 7
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
R. W. Cramer Co., P.O. Box 25, Centerbrook, Ct.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. (*p. 64*)
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Haydon Mfg. Co., 245 E. Elm St., Torrington, Ct.
International Register Co., 2620 W. Washington Blvd., Chicago 12, Ill.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Miller Harris Instr. Co., 836 N. 4th St., Milwaukee 3, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Reliance Automatic Lighting Co., 1927 Mead St., Racine, Wis.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
M. H. Rhodes, Inc., 20 Barholomew, Hartford, Ct.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Sangamo Elec. Co., Springfield, Ill.
Tagliabue Instrument Div., Weston Electrical Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

TIN

Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
Federated Metals Div., American Smelting & Refining Co., 120 B'way, N.Y.C. 5
National Lead Co., 111 Broadway, N.Y.C. 6
Northwest Lead Co., 2700-6th Ave., S.W., Seattle 4, Wash.

TIN PLATE

Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bethlehem Steel Co., Bethlehem, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
United States Steel Co., Subsidiary of U. S. Steel Corp., 525 Wm Penn Place Bldg., Pittsburgh 30, Pa.
Weirton Steel Co., Weirton, W. Va.
Wheeling Steel Corp., Wheeling, W. Va.
Youngstown Sheet & Tube Co., Youngstown, O.

TRACERS (See LEAK INDICATORS)

TRACK, SLIDING DOOR

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

TRANSFORMERS

Acme Elec. Corp., Cuba, N.Y.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American Transformer Co., 172 Emmet St., Newark 5, N.J.
Barber-Colman Co., Rockford, Ill.
Better Coils, Inc., Goodland, Ind.
Dean W. Davis & Co., 3418 N. Milwaukee Ave., Chicago 41, Ill.
Eisler Engrg. Co., Inc., 740 S. 13th St., Newark 3, N.J.
Esterline-Angus Co., Inc., P.O. Box 596, Indpls. 6, Ind.
Forbes & Myers, 173 Union St., Worcester, Mass.
France Mfg. Co., 10325 Berea Rd., Cleveland 2, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.J.
Jefferson Elec. Co., 25 Ave. & Madison St., Bellwood, Ill
Rex Welder & Engrg. Co., 1316 E. 10th St., Kansas City 6, Mo.
Wagner Elec. Corp., 6400 Plymouth Ave., St. Louis 14 Mo.

TRAPS, REFRIGERATOR DRAIN

Kason Hardware Corp., 127 Wallabout St., Brooklyn 6 N.Y.

TRAPS, STEAM (See STEAM TRAPS)

TRAPS, WATER, FOR AIR LINES

Armstrong Machine Wks., 831 Maple St., Three Rivers, Mich. (*p. 160*)
Climax Controls Div., Black Sivalls & Bryson, Inc., Box 1529, Tulsa 1, Okla.
De Vilbiss Co., 300 Phillips Ave., Toledo 1, O.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
W. H. Nicholson & Co., 209 Oregon St., Wilkes-Barre, Pa.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 (*p. 167*)
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.

TRAYS, DEFROSTING

General Industries Co., Olive & Taylor Sts., Elyria, O
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Premium Plastics, Inc. 627 Salem Ave., Dayton 6, O.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky
Sneath Glass Co., Hartford City, Ind.
Amos Thompson Corp., Edinburgh, Ind.

TRAYS, DISPLAY CASE

Bettinger Enamel Corp., Metal Fabricating Div., Waltham, Mass.
Gustar Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Ingram-Richardson Co., 32nd St., Beaver Falls, Pa.
McCray Refrigerator Co., Kendallville, Ind.
Premium Plastics. Inc., 627 Salem Ave., Dayton 6, O.
Amos Thompson Corp., Edinburgh, Ind.

TRAYS, ICE CUBE (See ICE CUBE TRAYS)

TRAYS, WIRE (See SHELVES; also WIRE FORMING)

TRIM, BREAKER (See BREAKER STRIPS)

TRIM, CABINET (See also NAME PLATES)

American Emblem Co., Inc., P.O. Box 116-H, Utica 1, N.Y.
Brasco Mfg. Co., Harvey, Ill.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y

(Continued)

TRIM, CABINET (Continued)

Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich.
Greene Mfg. Co., Inc., 1028 Douglas Ave., Racine, Wis.
Kent Plastics, 1528 N. Fulton Ave., Evansville, Ind.
John Lees Div., Serrick Corp., Muncie, Ind.
National Lock Co., Rockford, Ill.
Rigidized Metals Corp., 658 Ohio St., Buffalo 3, N.Y.
Sandee Mfg. Co., 5050 W. Foster Ave., Chicago 30, Ill.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stanley Wks., 195 Lake St., New Britain, Ct.
Amos Thompson Corp., Edinburg, Ind.
U. S. Gypsum Co , 300 W. Adams St., Chicago 6, Ill.
R. D. Werner Co., Inc., 295-5th Ave., N.Y.C. 16

TROLLEYS, OVERHEAD CONVEYOR (See CONVEYORS)

TRUCK & TRAILER BODIES, REFRIGERATED

Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. *(p. 152)*
Divco Corp., 22000 Hoover Rd., Detroit 5, Mich.
Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.
Fruehauf Trailer Co., 10925 Harper Ave., Detroit, Mich.
Robbins & Burke, Inc., 29 Lansdowne, Cambridge, Mass
Stokes Co., 131st & Morgan Sts., Blue Island, Ill.

TRUCK & TRAILER REFRIGERATION SYSTEMS

Advance Mfg. Co., Inc., 2700 Buchanan St., Detroit 8, Mich.
American Mfg. Co., Inc., P.O. Box, 989, Montgomery 2, Ala.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Cold Control, Inc., 111 Broadway, N.Y.C. 6
Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. *(p. 152)*
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
Franklin Body & Equipment Corp., 1042 Dean St., Brooklyn 16, N.Y.
Frick Co., Waynesboro, Pa. *(p. 51)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
International Harvester Co., 180 No. Michigan Ave., Chicago 1, Ill
Keco Industries, Inc., 2519 Gilbert Ave., Cincinnati 6, O.
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 153)*
Propane Development Corp., 41 Murray St., N.Y.C. 7
Stokes Co., 131st & Morgan Sts., Blue Island, Ill.
Trane Co., La Crosse, Wis. *(p. 118)*
U. S. Thermo Control Co., 44 S. 12th St., Minneapolis 4, Minn.

TRUCK CASTERS

Bossick Co., 437 Howard Ave., Bridgeport 2, Ct.
Colson Corp., Cedar St., Elyria, O.
Darnell Corp., Ltd., P.O. Box 4027, Sta. B., Long Beach 4, Cal.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Menasha Wood Split Pulley Co., 674 Tayco St., Menasha, Wis.
Nutting Truck & Caster Co., Inc., 1125 W. Division St., Fairbault, Minn.

TRUCK PLATES

Dean Products, Inc., 1042 Dean St., Brooklyn 16, N.Y. (p. *152*)
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 153)*

TRUCKS, MATERIAL HANDLING

Buda Co., 15th & Commercial Ave., Harvey, Ill.
Colson Corp., Cedar St., Elyria, O.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Lift Trucks, Inc., 2425 Spring Grove Ave., Cincinnati 14, O.
Menasha Wood Split Pulley Co., 674 Tayco St., Menasha, Wis.
Nutting Truck & Caster Co., Inc., 1125 W. Division St., Faribault, Minn.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.

TUBE BENDING (See also PIPE COILS, BENDS & BENDING)

Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
American Brass Co., Waterbury 88, Ct. *(p. 189)*
American District Steam Co., Bryant St., N., Tonawanda, N.Y.
Betz Corp., 445 State St., Hammond, Ind.
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
C. F. Moores Co., Inc., Ivy Hill Rd., Wyndmoor, Phila 18, Pa.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
V. E. Sprouse Co., Inc., 1804-22nd St., Columbus, Ind.
York Corp., York, Pa. *(p. 52)*

TUBE ICE MACHINERY (See ICE MAKING MACHINES)

TUBES & TUBING, ALLOY

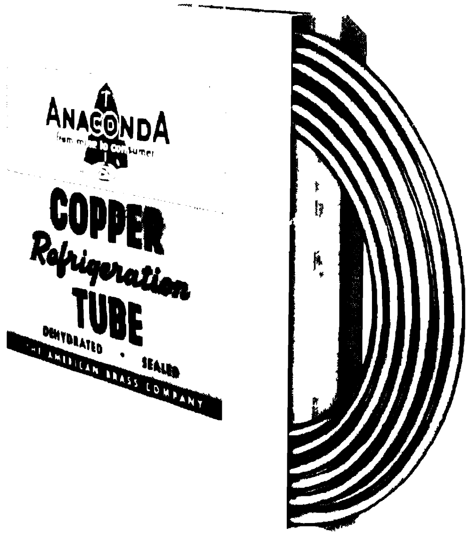
American Brass Co., Waterbury 88, Ct. *(p. 189)*
Babcock & Wilcox Co., 85 Liberty St. N.Y C. 6
Chase Brass & Copper Co , 236 Grand St., Waterbury 91, Ct.
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
National Tube Co., U. S. Steel Corp., Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Ohio Seamless Tube Co., 132 W. Main St., Shelby, O.
Pacific Tube Co., 5710 Smithway St., Los Angeles 22, Cal.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 2100 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wallingford Steel Co., Wallingford, Ct.

TUBES & TUBING, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
American Brass Co., Waterbury 88, Ct. *(p. 189)*
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Colonial Alloys Co., Ridge Ave. & Crawford St., Phila. 29, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.

(Continued)

ANACONDA PRODUCTS FOR THE REFRIGERATION INDUSTRY



ANACONDA COPPER REFRIGERATION TUBES have qualities that assure a dependable ready and easy to use refrigeration tube

they are dehydrated to eliminate troublesome moisture

they come to you in single coil cartons with unusually clean and bright inside surfaces

they are sealed to protect the inside of the tube from the point of manufacture to the point of installation

they are uniformly soft easy to bend and may be readily flared without cracking

Anaconda Refrigeration Tubes are 99 9% pure copper made to A S T M Specification B68 They are available in all standard sizes up to and including ¾" O D in 50 foot coils Longer lengths on special order

PRECISION-MADE TUBES in Copper Brass Bronze and Copper Nickel Alloys in sizes from 0 15" O D to 1" O D with wall thickness down to 004" round square or irregularly shaped

RESTRICTOR TUBE

FORMED TUBE PARTS

HARD COPPER TUBE CUT TO LENGTH

COPPER WATER TUBE IN COILS AND STRAIGHT LENGTHS—TYPES K AND L

TUBE FITTINGS

VIBRATION ELIMINATORS

CHARGING HOSE

FLEXIBLE REFRIGERATION TUBING CONDUIT

DIE PRESSED FORGINGS

COPPER BRASS BRONZE IN SHEETS WIRE, RODS, TUBES AND SPECIAL SHAPES

THE AMERICAN BRASS COMPANY, General Offices Waterbury 20, Connecticut
In Canada New Toronto Ontario

TUBES & TUBING, ALUMINUM (Continued)

Linderme Tube Co., 1500 E. 219th St., Cleveland 17, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

TUBES & TUBING, BRASS & BRONZE

American Brass Co., Waterbury 88, Ct. (*p. 189*)
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburgh 19, Pa.
Lewin Metals Div., Lewin-Mathes Co., 1111 Chouteau, St. Louis, Mo.
Linderme Tube Co., 1500 E. 219th St., Cleveland 17, O.
MacKenzie-Walton Co., 480 Pawtucket Ave., Pawtucket, R.I.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Penn Brass & Copper Co., Powell Ave., Erie, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Shenango-Penn Mold Co., Dover, O.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1850 Guardian Bldg., Detroit 26, Mich.

TUBES & TUBING, CAPILLARY

American Brass Co., Waterbury 88, Ct. (*p. 189*)
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. (*p. 193*)
MacKenzie-Walton Co., 480 Pawtucket Ave., Pawtucket, R.I.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Penn Brass & Copper Co., Powell Ave., Erie, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Superior Tube Co., 2100 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1850 Guardian Bldg., Detroit 26, Mich.

TUBES & TUBING, CAPILLARY (ASSEMBLIES)

Allin Mfg. Co., 1153 W. Grand Ave., Chicago, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Northwest Automatic Products Corp., Linndale & Linden, Minneapolis, Minn.
Wabash Mfg. Co., 2300 South Western Ave., Chicago 8, Ill.

TUBES & TUBING, COPPER

American Brass Co., Waterbury 88, Ct. (*p. 189*)
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburgh 19, Pa.
Kensico Tube Co., Mt. Kisco N.Y.
Lewin Metals Div., Lewin-Mathes Co., 1111 Chouteau, St. Louis, Mo.
Linderme Tube Co., 1500 E. 219th St., Cleveland 17, O.
MacKenzie-Walton Co., 480 Pawtucket Ave., Pawtucket, R.I.
Mueller Brass Co., Port Huron, Mich.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Penn Brass & Copper Co., Powell Ave., Erie, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Viking Copper Tube Co., 16700 St. Clair Ave., Cleveland 10, O.
Wayne Brass Co., Inc., 230 W. Main St., Fort Wayne, Ind.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1850 Guardian Bldg., Detroit 26, Mich.

TUBES & TUBING, FIBRE (See FIBRE, SHEET, ROD & TUBE)

TUBES & TUBING, INCONEL, MONEL, NICKEL

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. (*p. 193*)
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Driver-Harris Co., Harrison, N.J.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
MacKenzie-Walton Co., 480 Pawtucket Ave., Pawtucket, R.I.
National Tube Co., U. S. Steel Corp., Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pacific Tube Co., 5710 Smithway St., Los Angeles 22, Cal.
Superior Tube Co., 2100 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

TUBES & TUBING, RUBBER

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Flexible Tubing Corp., New Whitfield St., Guilford, Ct.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
New York Belting & Packing Co., 1 Market St., Passaic N.J.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

TUBES & TUBING, SPONGE RUBBER

General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 171*)
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. (*p. 85*)
Sponge Rubber Products Co., 1600 Derby Place, Shelton, Ct.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

TUBES & TUBING, STEEL

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. (*p. 193*)
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.

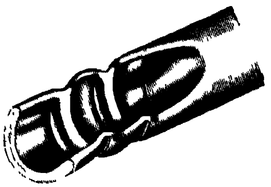
(*Continued*)

REVERE DRYSEAL COPPER REFRIGERATION TUBE

HANDY Package

Revere Dryseal Tube comes in 50-foot double layer flat coils, packed in individual cartons. These sturdy cartons protect the tube, keep it clean, may be conveniently stacked for storage, and are easier to identify and to unpack.

PROTECTIVE Seal



Revere Dryseal Tube has a double-groove mechanical seal. It is compact enough to pass through any opening large enough for the tube itself. It permanently keeps the interior of the tube clean and bone dry.

Standard SIZES

Revere Dryseal Tube comes in economical dimensional standards as follows:

Size O.D. in inches	Wall Gauge in inches	Pounds per Linear Foot	Approx. Weight per 50-ft. Coil
1/8	.030	.0347	1.74
3/16	.030	.0575	2.88
1/4	.030	.0804	4.02
5/16	.032	.109	5.45
3/8	.032	.134	6.70
1/2	.032	.182	9.10
5/8	.035	.251	12.55
3/4	.035	.305	15.25

Revere Dryseal is the Standard of Quality

Revere Dryseal Tube is dead soft, so that you can bend it easily . . . and the ends won't split when flared. Exterior and interior surfaces are clean, smooth, and free from mechanical defects. It is made of deoxidized copper (99.9% pure) and is kept oxide-free by special processing methods.

Revere Dryseal Copper Refrigeration Tube is available from leading distributors throughout the country. Revere also manufactures aluminum refrigeration tube.

REVERE COPPER AND BRASS INCORPORATED

Founded by Paul Revere in 1801
230 Park Avenue, New York 17, New York
Mills: Baltimore, Md.; Chicago, Ill.; Detroit, Mich.; Los Angeles and Riverside, Calif.;
Clinton, Ill.; New Bedford, Mass.; Rome, N.Y.—
Sales Offices in Principal Cities, Distributors Everywhere

TUBES & TUBING, STEEL (Continued)

Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Michigan Seamless Tube Co., S. Lyon, Mich.
Michigan Steel Tube Products Co., 9450 Buffalo St., Detroit 12, Mich.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P. O. Box 266, Pittsburgh, Pa.
Ohio Seamless Tube Co., 132 W. Main St., Shelby, O.
Pacific Tube Co., 5710 Smithway St., Los Angeles 22, Cal.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 2100 Germantown Ave., Norristown, Pa.

TUBES & TUBING, STEEL, STAINLESS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Columbia Steel & Shafting Co., P.O. Box 1557, Pittsburgh 30, Pa.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pacific Tube Co., 5710 Smithway St., Los Angeles 22, Cal.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 2100 Germantown Ave., Norristown, Pa.
Wallace Tube Co., Sub. of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.
Wallingford Steel Co., Wallingford, Ct.
Wooster Brass Co., 1415 E. Bowman St., Wooster, O.

TUBES, BOILER & CONDENSER

American Brass Co., Waterbury 88, Ct. (*p. 189*)
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Columbia Steel & Shafting Co., P.O. Box 1577, Pittsburgh 30, Pa.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Ohio Seamless Tube Co., 132 W. Main St., Shelby, O.
Pacific Tube Co., 5710 Smithway St., Los Angeles 22, Cal.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 (*p. 191*)
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Superior Tube Co., 2100 Germantown Ave., Norristown, Pa.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUBING, BEER

Aeroquip Corp., 300 S. East Ave., Jackson, Mich.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. (*p. 193*)
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 45, Wis.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7 R.I

TUBING, BLOCK TIN

Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
National Lead Co., 111 Broadway, N.Y.C. 6
Northwest Lead Co., 2700-16th Ave., S.W., Seattle 4, Wash.

TUBING, FINNED (See FINNED TUBING)

TUBING, FLEXIBLE METAL (See also CHARGING LINES; also FLEXIBLE CONNECTIONS)

Aeroquip Corp., 300 S. East Ave., Jackson, Mich.
Allied Metal Hose, 37-46 9th St., Long Island City, N.Y.
American Brass Co., Waterbury 88, Ct. (*p. 189*)
Atlantic Metal Hose Co., Inc., 123 W. 64th St., N.Y.C. 23
Chicago Metal Hose Corp., Maywood, Ill.
Clifford Mfg. Co., Div. of Std. Thomson Co., Grove St., Waltham, Mass.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Federal Metal Hose Corp., 277 Military Rd., Buffalo 7, N.Y.
Flexible Tubing Corp., New Whitfield St., Guilford, Ct.
Johnson Metal Hose Co., 226 Mill St., Waterbury 88, Ct.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.Y.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y
Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
Uniflex Metal Hose Co., 52 Rubber Ave., Naugatuck, Ct.

TUBING, LEAD

Bridgeport Thermostat Co., Inc., 1225 Connecticut Ave. Bridgeport 1, Ct.
Division Lead Co., 836 Kinzie St., Chicago 22, Ill.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
National Lead Co., 111 Broadway, N.Y.C. 6
Northwest Lead Co., 2700-16th Ave., S.W., Seattle 4, Wash.

TUBING, PREFABRICATED & TUBULAR PARTS

Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
W. R. Ames Co., 150 Hooper St., San Francisco 7, Cal.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. (*p. 193*)
Capitol Mfg. & Supply Co., 153 W. Fulton St., Columbus, O.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Michigan Steel Tube Products Co., 9450 Buffalo St., Detroit 12, Mich.
Mueller Brass Co., Port Huron, Mich.
A. B. Murray Co., Inc., 630 Green Lane, Elizabeth, N.J.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Ohio Seamless Tube Co., 132 W. Main St., Shelby, O.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.

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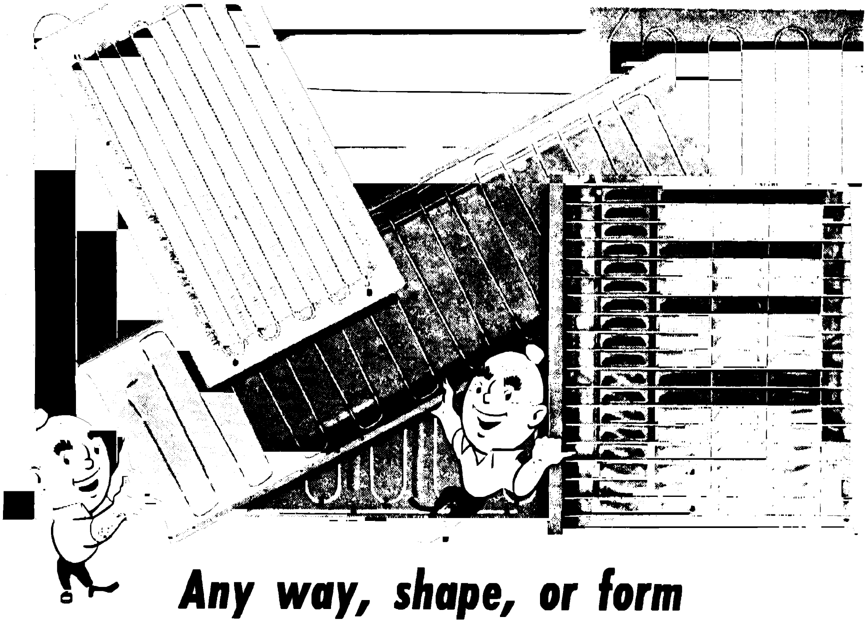


Any way, shape, or form

It's Bundyweld for better refrigeration!

In coils or condensers. In compressors, evaporators, or refrigerant lines. Whatever your application, Bundyweld spells more advantages than any other tubing. It's double-walled from a single strip, copper-brazed through 360° of wall contact . . . stronger, lighter-weight, easier-forming, easier-handling.

Too, Bundyweld's double walls mean thinner walls for faster heat conductivity; finer, more dependable performance for the life of your refrigerators, home or commercial freezers, display cases, refrigerated cars, whatever you make that calls for refrigeration tubing.

Foiled by a coil?

Then bring that tricky bend to Bundy, where exclusive, patented bending machines can roll out formed tubing parts to perfection, ready for your assembly lines. When a given bending fixture doesn't exist, Bundy engineers will often design a new one to lick the knottiest bending problems. For the complete Bundyweld story, write: ***Bundy Tubing Company, Detroit 14, Mich.***

Bundyweld Tubing®

DOUBLE-WALLED FROM A SINGLE STRIP

NOTE the exclusive patented Bundyweld beveled edge, which affords a smoother joint, absence of bead and less chance for any leakage.

SIZES UP TO ⅝" O. D.

TUBING, PREFABRICATED (Continued)

Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUNNEL FREEZERS (See FREEZERS)

TURBINES (See STEAM TURBINES)

TURBO COMPRESSORS (See also COMPRESSORS, CENTRIFUGAL; also CONDENSING UNITS, CENTRIFUGAL)

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Elliott Co., Jeannette, Pa.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind
Spencer Turbine Co., Hartford 6, Ct.
Trane Co., La Crosse, Wis. *(p. 118)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p 144)*
York Corp., York, Pa. *(p 52)*

TURNBUCKLES

Bethlehem Steel Co., Bethlehem, Pa.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
U. S. Expansion Bolt Co., York, Pa.

TURNINGS (See SCREW MACHINE PARTS; also THREADED SPECIALTIES)

TWO TEMPERATURE VALVES, SNAP ACTION (See also SUCTION PRESSURE REGULATING VALVES)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 99)*
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 149)*
A-P Controls Corp., 2450 N. 32nd St., Milwaukee 10, Wis.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 46)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p 17)*
Mueller Brass Co., 1925 Lapeer Ave., Port Huron, Mich.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p 89)*

ULTRA-VIOLET RADIATION (See LAMPS, ABCTERICIDAL)

UNDERCOATERS (See COATINGS & COMPOUNDS; also FINISHES)

UNIONS, PIPE (See FITTINGS, PIPE)

UNIT COOLERS ABOVE 32°—CEILING MOUNTED

(A—Ammonia; B—Other refrigerants)

(A,B) Advanced Engrg. Co., 2646 W. Fond du Lac Ave, Milwaukee, Wis.
(B) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B) Baker Refrigeration Corp., S. Windham, Me. *(p. 68)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(A,B) Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(A) Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
(A,B) Conditionaire Unit Co., Div. of Howe Ice Machine Co., 2815 Montrose Ave., Chicago 18, Ill. *(p. 198)*
(A) Creamery Package Mfg. Co., 1243 W. Washingington Blvd., Chicago 7. Ill. *(p. 49)*
(B) Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. *(p. 7)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(B) Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(B) Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y. *(p. 59)*
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 29)*
(B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(A,B) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
(B) Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo.
(A,B) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill
(A,B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. *(p. 10)*
(A,B) King Co., 902 N. Cedar St., Owatonna, Minn.
(A) Kohlenberger Engrg. Corp. 1600 W. Commonweath, Fullerton, Cal. *(p. 39)*
(A,B) Kramer Trenton Co., Olden & Bruening Aves, Trenton 5, N.J. *(p. 195)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(B) McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. *(p. 55)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
(A,B) D. J. Murray Mfg. Co., 1024-3rd St., Wausau, Wis
(A,B) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p 97)*
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p 201)*
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(A,B) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(A,B) Super Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Trane Co., La Crosse, Wis. *(p. 118)*
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E. at 33rd St., Minneapolis 14, Minn. *(p 87)*
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 88)*
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 52)*

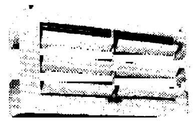
UNIT COOLERS ABOVE 32°—FLOOR MOUNTED

(A—Ammonia; B—Other refrigerants)

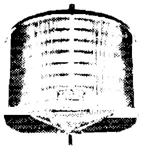
(A,B) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
(A.B) Baker Refrigration Corp., S. Windham, Me. *(p 68)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(A,B) Buffalo Forge Co., 217 Mortimer St., Buffalo 10, N.Y.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(B) Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.

(Continued)

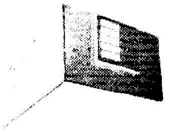
KRAMER PRODUCTS



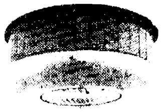
COOLMASTER
2,500 to 60,000 BTU's Per Hour



CURVETTE
700 to 20,000 BTU's Per Hour



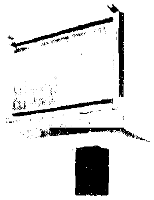
AIR CONDITIONING UNIT
3 to 10 Ton



RADIAL
2,500 to 23,000 BTU's Per Hour

UNICON
¼ to 10 Ton

COOLING TOWER
3 to 8 Ton



EVAPORATIVE CONDENSER
2 to 10 Ton

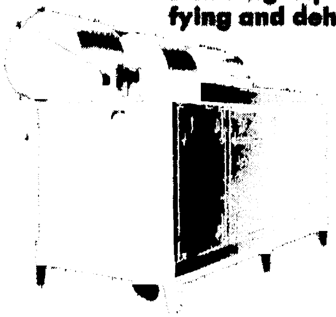
Only **THERMOBANK** by **KRAMER**

Keeps Coils Frost-Free *Automatically* at Any Temperature *without*...

LABOR
ATTENTION
ELECTRIC HEATERS
BRINE OR WATER SPRAYS

KRAMER TRENTON CO. • Trenton 5, N.J.

Commercial and industrial refrigeration equipment for temperatures above and below 34° F. Air conditioning equipment for cooling, heating, humidifying and dehumidifying.

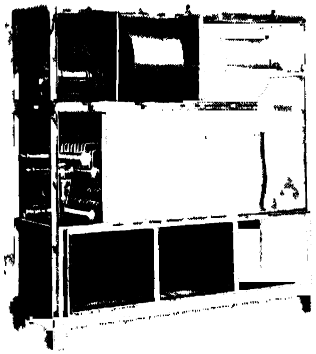


"Dri-Fan" Evaporative Condensers
"Dri-Fan" Cooling Towers

"Dri-Fan" Evaporative Condensers and Cooling Towers. Capacities from 3 to 150 tons.

Air Conditioning Units
Multizone Air Conditioners
Cooling and Heating Coils

Water Defrost Blower Units
Refrigeration Blower Coils
Ceiling and Floor Type Units



RECOLD Air Conditioners and Industrial Refrigeration Units. Capacities: Air Conditioners from 3 to 50 tons. Industrial Units from 1 to 15 tons.

RECOLD
TRADE-MARK REG.

Write for complete information today

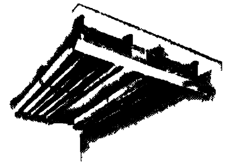
Refrigeration ENGINEERING, INC.

7260 EAST SLAUSON AVENUE • LOS ANGELES 22, CALIFORNIA

Refrigeration Economics Co., Inc.

1231 Tuscarawas St. E., Canton 2, Ohio

Recoy Products

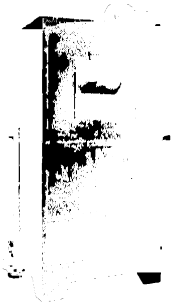


← **C. T. COILS**

Continuous tube down draft fin coils are still unsurpassed for meat coolers, or other products requiring high humidity and gentle air circulation

EVAPORATIVE CONDENSERS →

Evaporative condensers from 2 to 100 tons Brine spray cooling to 25 tons

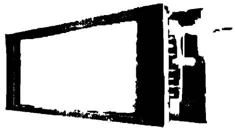


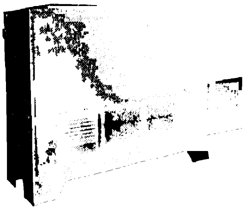
← **CEILING DIFFUSER**

Ceiling diffusers distribute the cooled air across the ceiling so the draft does not strike the products stored or occupants

C. F. COILS →

Continuous fin coils for unit coolers blast heaters air conditioning and condensers



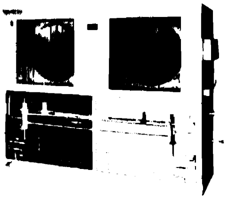


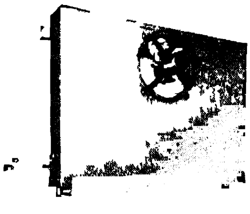
← **AIR CONDITIONING**

Air conditioning units of ceiling or floor type in all capacities for cooling heating, or both

AUTOMATIC DEFROST UNITS →

Complete ready for electric, liquid suction and hot gas connections One coil working always, both 98 per cent of time



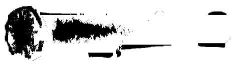


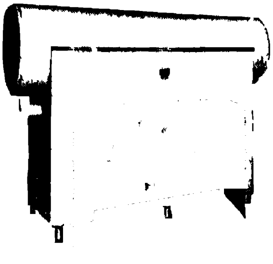
← **WALL UNITS**

Recoy "All Seasons" wall units provide a damper for deflecting the cold air down along the wall or out horizontally into the room thus providing proper air circulation for "All Seasons"

SHELL CONDENSER →

Shell and tube, also shell and fin coil condensers Both types have tubes arranged for cleaning with mechanical tube cleaner





← **WATER COOLING**

Self contained complete ice water and brine coolers complete with high and low sides, circulating pumps controls and insulation 3 to 30 hp

FLOOR UNITS →

Floor units with cooling surface exposed to view have a definite advantage over those with coils hidden Design permits water defrosting

UNIT COOLERS ABOVE 32° (Continued)

(A,B) Frick Co., Waynesboro, Pa. (*p. 51*)
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(A,B) Gay Engrg Co , 2730 E 11th St., Los Angeles 23, Cal.
(A,B) Howe Machine Co., 2825 Montrose Ave., Chicago 18, Ill. (*p. 198*)
(A) Industrial Mfg & Engrg Co , 3845 N Ravenswood Ave , Chicago 13, Ill
(A,B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo. (*p. 6*)
(A,B) King Co , 902 N. Cedar St., Owatonna, Minn
(A) Kohlenberger Engrg. Corp. 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
(A) Kramer Trenton Co., Olden & Breuning Aves. Trenton 5, N.J. (*p. 195*)
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. (*p. 41*)
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. (*p. 200*)
(A,B) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
(B) Read Machinery Div Standard Stoker Co , Inc., York, Pa.
(A,B) Refrigeration Appliances, Inc , 917 W. Lake St., Chicago 7, Ill. (*p. 201*)
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
(A,B) Rigidbilt, Inc , 2850 W. Fulton St., Chicago 12, Ill.
(A,B) Stewart Ice Machine Co , 1282 W. 1st St., Pomona, Cal.
(A,B) Super Cold Corp 1020 E 59th St Los Angeles 1, Cal.
(B) Tenney Engrg , Inc , 26 Ave B, Newark 5, N.J.
(A,B) Trane Co., La Crosse, Wis. (*p. 118*)
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p. 37*)
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J (*p. 144*)
(A,B) XL Refrigerating Co., 1834 W. 59th St , Chicago 36, Ill.
(A,B) York Corp., York, Pa. (*p. 52*)

UNIT COOLERS ABOVE 32°—WALL MOUNTED

(A—Ammonia; B—Other refrigerants)

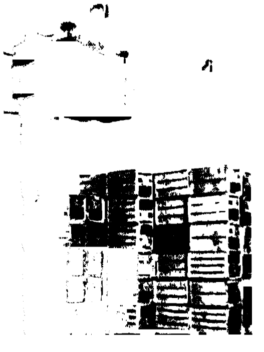
(B) American Coils Co , 25 Lexington St. Newark 5, N.J
(A,B) Baker Refrigeration Corp., S. Windham, Me. (*p. 65*)
(B) Betz Corp , 445 State St , Hammond, Ind.
(A,B) Buffalo Forge Co 217 Mortimer St , Buffalo, N Y
(B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. (*p. 40*)
(B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. (*p. 49*)
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(B) Evans Mfg Corp , 460 S 10th Ave , Mt Vernon, N Y.
(B) Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y. (*p. 59*)
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(B) Hussmann Refrigerator Co., Inc., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 83*)
(B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 7, Mo. (*p. 10*)
(A) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
(A,B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
(B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. (*p. 41*)
(B) McCord Corp., Riopelle & E. Grand Blvd., Detroit 11, Mich. (*p. 55*)
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. (*p. 199*)
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. (*p. 200*)
(A,B) D. J. Murray Mfg. Co., 1024-3rd St., Wausau, Wis.
(A,B) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 (*p. 97*)
(B) Reese & Long Refrigeration Products, Inc., 1891 Park Ave N Y C 35
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. (*p. 201*)
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. (*p. 196*)
(A,B) Rempe Co , 310 N. Sacramento Blvd , Chicago 12, Ill.
(B) Sterling Mfg Co , 2523 Farnam St , Omaha, Neb
(A,B) Stewart Ice Machine Co., 1282 W 1st St , Pomona, Cal
(A,B) Super Cold Corp , 1020 E 59th St , Los Angeles 1, Cal
(B) Tenney Engrg , Inc., 26 Ave B, Newark 5, N J.
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn. (*p 37*)
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
(A,B) XL Refrigerating Co., 1834 W. 59th St , Chicago 36, Ill.
York Corp, York, Pa (*p 52*)

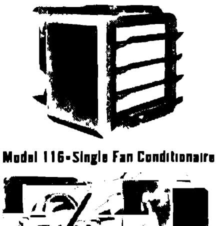
UNIT COOLERS BELOW 32°—CEILING MOUNTED

(A—Ammonia; B—Other refrigerants)
(a—Reverse cycle; b—Electric; c—Brine spray; d—Water; e—Glycol; f—Warm air; g—Others)

(B b) American Coils Co , 25 Lexington St., Newark 5, N.J.
(A,B:a,c,e) Baker Refrigeration Corp., S. Windham, Me. (*p. 63*)
(B a,b,d) Betz Corp , 445 State St , Hammond, Ind

(Continued)





CONDITIONAIRE UNIT COOLERS
Ceiling Type, for ALL Commercial Uses

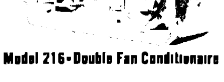
• Use Ammonia, Methyl, Freon, Brine or Water with a Conditionaire. Its all steel, welded, hot galvanized coil is suitable for any refrigerant. Other Conditionaire features: Sweat-proof, corrosion-free housing; silent motor; adjustable deflectors, any angle; thermo or float valve control. Sizes for every need. *Write for detailed literature!*

Conditionaire installation • A. C. Hipp Co , Brownsville, Tex.

CONDITIONAIRE UNIT CO.
Sales Division — Howe Ice Machine Co.
2821 MONTROSE AVENUE-CHICAGO 18, ILLINOIS

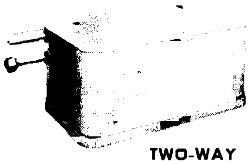
Model 116 • Single Fan Conditionaire

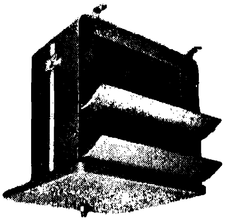
Model 216 • Double Fan Conditionaire

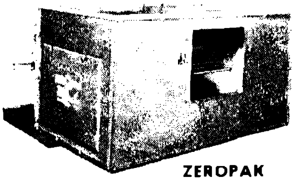


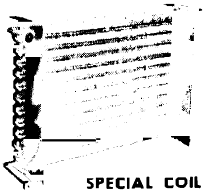
McQuay PROVEN PRODUCTS

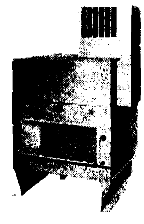
MANUFACTURERS OF EQUIPMENT FOR ALL PHASES OF AIR CONDITIONING


TWO-WAY UNIT COOLER


PACEMAKER UNIT COOLER


ZEROPAK Model "L"


SPECIAL COIL


EVAPORATIVE CONDENSER

Two-Way Unit Coolers. A small compact unit designed for reach-ins, back bars, and direct draw bars where space is limited. All connections are on the outside for ease of valve mounting.

Radial Unit Coolers. A unit styled for use where minimum depth and height are required. It may be used for either refrigeration or comfort cooling applications.

Pacemaker Unit Coolers. An ideal unit for walk-in coolers, back bars, etc. Constructed for long life and service free operation.

Panel Type Chillators. A modern attractive unit especially adaptable for wall mounting in reach-in cabinets and small low ceiling walk-ins.

Model "X" Zeropak. Developed primarily for locker plants as an efficient cooler and sharp freezer combined. This unit has been found very suitable for all types of low temperature storage.

Model "L" Zeropak. The leader in the low temperature field for locker plants, meat freezing rooms, and ice cream hardening rooms. Types "C" and "CA" storage coolers for extra heavy product loads in large rooms above 35°F are also available.

Multi Louvre Coils. For use where high humidity and slow uniform air movement is required, as in floral boxes or cheese rooms.

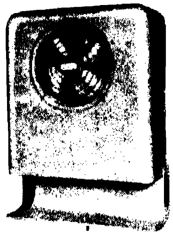
Special Coils. All types of special coils such as heavy duty evaporators and condensers for refrigerated trucks, rail cars, etc., as well as 5/8" tube evaporators and condensers for light tonnage applications.

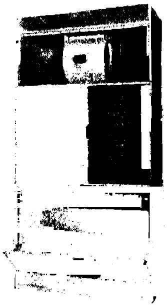
Evaporative Condensers. Available in 3½, 6 and 12 ton sizes. Copper coils of all prime surface, unique pan type water distributor and automatic water sweetening provide for economy of operation. Units can be floor or ceiling mounted with horizontal or vertical air discharge.

RH Air Conditioners. This unit may be installed in the conditioned space or in an adjacent area with or without duct work. In addition to the freon coil they are available with water and steam coils for year round application.

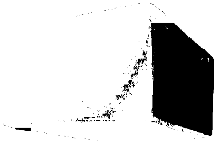
Descriptive bulletins of all McQuay products available upon request.

RADIAL UNIT COOLER


CHILLATOR (Panel Type)


ZEROPAK Model "X"

MULTI-LOUVRE COIL

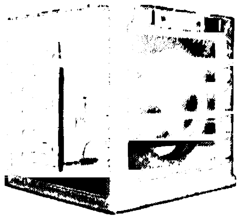

RH AIR CONDITIONER

McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn.

Marlo

Industrial Coolers—Unit Coolers—Evaporative Condensers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils—Cooling Towers—Diesel Engine and Oil Evaporative Coolers

Quality Heat Transfer Equipment Since 1925



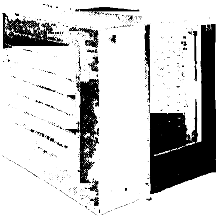
UNIT COOLER—DUC

UNIT COOLERS—UC AND DUC

11 unit sizes (75 to 4100 cfm Blow through and Pull through types *Write for Bulletins 392 and 412*

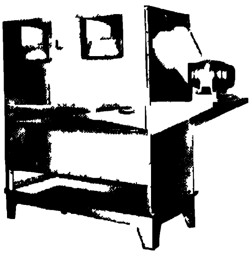
ELECTRIC DEFROST UNITS

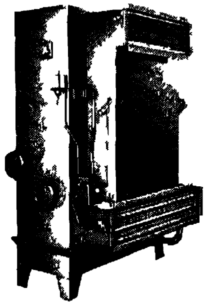
(U S Patent 2266373) 7 sizes Ammonia or Freon to 2 tons at 12 deg TD *Bulletin 405*



BLAST COILS

Air conditioning Industrial Refrigeration Heating Any material All refrigerants Every application Pressure expanded staggered tube continuous plate type fin *Bulletin 396*





INDUSTRIAL COOLERS

15 unit sizes—1000 to 26,400 cfm—Floor type Dry Coil and Spray types *Bulletin 403*

AIR CONDITIONING UNITS

Cooling — Heating — Dehumidifying — Humidifying 14 sizes—1 to 150 tons 640 to 57,200 cfm Ceiling suspended floor and multizone types *Bulletin 402 and 409 MZ*

EVAPORATIVE CONDENSERS

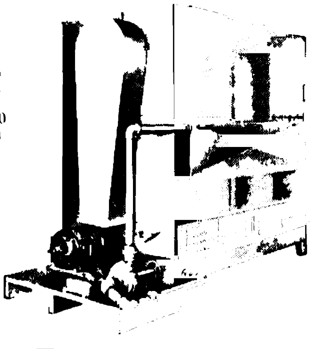
3 to 100 tons– All refrigerant —All prime surface coils No fans Quiet—Motor Unit drive Indoor or outdoor unit Durable construction *Bulletin 404*

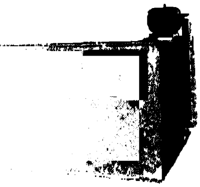
COOLING TOWERS

Induced Draft Wetted Deck Surface—Water Spray Compact in space and weight Outdoor Indoor 3 to 100 tons Sectional construction *Write for Bulletin 406*

EVAPORATIVE DIESEL COOLER

2 HP to 4000 HP





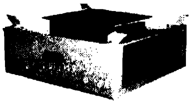
MARLO = HEAT TRANSFER
Since 1925

COIL CO. • 6135 Manchester Rd. • St. Louis 10, Mo.

KRACK *QUALITY*

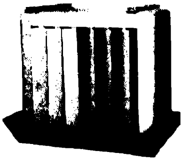
REFRIGERATION AND AIR CONDITIONING EQUIPMENT

STRAT-E-FEX



Low velocity ceiling unit with 360° air discharge. Built-in heat exchanger. 8 sizes for applications requiring humidity and low air velocity. Range: 150-1300 BTU/HR/1°FTD.

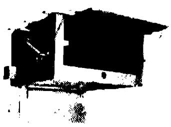
FAN-E-FEX



Popular priced all-purpose ceiling unit cooler with adjustable louvres and full size heat exchanger. For freon, ammonia, or brine. 12 sizes. Range: 60-1400 BTU per hour 1°FTD.

OTHER HIGH TEMPERATURE UNITS: Junior models for specific fixture applications. FLO-E-FEX: Panel cooler with patented pyramid deflector. DRAFT-E-FEX: Down draft unit cooler for limited space. MAGN-E-FLEX: Heavy duty twin fan model for big tough jobs. SUR-E-FEX: Custom made gravity coils.

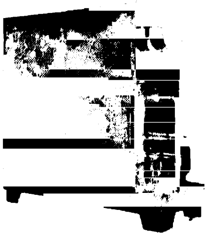
FREEZ-E-FEX



The original automatic electric defrost low temperature unit for storage or fast freezing. Completely insulated housing. Range: ¾-3 ton. For use with freon or ammonia.

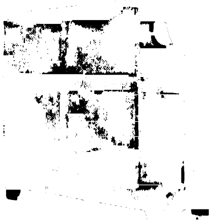
EVAPORATIVE CONDENSER

Floor type, 9 sizes, 5-50 tons. 12 ga. channel formed welded steel construction, hot dip galvanized *after fabrication*. Intake eliminators positively contain sump splash. Demountable sections facilitate handling. Turnover blower section for top, left, or right air discharge. Ceiling type in smaller sizes.



COOLING TOWER

Floor type, 8 sizes, 5-40 tons. 12 ga. channel formed welded steel construction hot dip galvanized *after fabrication*. Demountable sections, turnover blower section and intake eliminators. Removable doors provide easy access to interior. Separate motors for pump and blowers.



BLO-E-FEX

Heavy duty floor type industrial blower unit for high or low temperature applications. 12 ga. channel formed steel housing. Sectional turnover blower section. Directional hoods with adjustable louvres optional. Accessible interior. For freon, ammonia, or brine. 12 sizes. Range: 1-15 tons.

OTHER AIR CONDITIONING EQUIPMENT: BREEZ-E-FEX, Console comfort cooler (¾-1½ tons). COMF-E-FEX, ceiling comfort cooler (¾-3 tons). Also ceiling (5-25 tons) and floor (5-50) central station air conditioners as well as custom made air conditioning coils.

- ***Scientifically Engineered*** • ***Precision Made*** • ***Balanced Performance***

Specifications and Data Sheets on any Unit Available on Request

REFRIGERATION APPLIANCES, INC.

901-923 W. Lake St., Chicago 7, Illinois, U.S.A.

Pioneer Manufacturers of Quality Heat Transfer Equipment

UNIT COOLERS BELOW 32° (Continued)

(A,B:f) Buffalo Forge Co., 217 Mortimer St., Buffalo, N.Y.
(A,B:c,e.) Bush Mfg. Co., 179 South St., W. Hartford 10: Ct. *(p. 40)*
(A,B:d) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(A,B:d) Conditionaire Unit Co., Div. of Howe Ice Machine Co., 2815 Montrose Ave., Chicago 18, Ill. *(p. 198)*
(A:a,c) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
(B:d) Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. *(p. 7)*
(A,B:a,b,c,d) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Evans Mfg. Corp., 460 S. 10th St., Mt. Vernon, N.Y.
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B:a,b,c,d) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(A,B:a,d) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
(B:d) Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 83)*
(AB,f) King Co., 902 N. Cedar St., Owatonna, Minn.
(A:a,c) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
(A,B:g) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 195)*
(A,B:d) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 41)*
(A,B:b) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(A.B:b,c) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
(A,B) D. J. Murray Mfg. Co., 1024-3rd Ave., Wausau, Wis.
(A,B:e) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
(A,B:a,b,c,d) Reco Products Div., Refrigeration Engrg Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B:b,d) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
(A,B:a,b,d) Refrigeration Economics Co., Inc., 1231 Tuscarawas St., Canton 4, O. *(p. 197)*
(A,B:d) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(A,B:a) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(A,B:c,d) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
(A,B:a,b,c,d,e,f,g) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(g) Surface Combustion Corp., 2375 Dorr St., Toledo 1, O.
(B:d,f,g.) Tenney Engrg., Inc., 26 Ave. B. Newark 5, N.J.
(A,B:a,c) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
(c,e) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A,B:a,c,d,f) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

UNIT COOLERS BELOW 32°—FLOOR MOUNTED

(A—Ammonia; B—Other refrigerants)
(a—Reverse cycle; b—Electric; c—Brine spray; d—Water; e—Glycol; f—Warm air; g—Other)

(A,B:c) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
(A,B:a,c,e) Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
(B:d) Betz Corp., 445 State St., Hammond, Ind.
(A,B:f) Buffalo Co., 217 Mortimer St., Buffalo, N.Y.
(A,B:d) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 40)*
(A,B:d) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(A:c) Creamery Package Mfg., Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
(A,B:a,b,c,d) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B:a,c,d,f) Frick Co., Waynesboro, Pa. *(p. 51)*
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B:a,b,c,d) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
A,B:a,d) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 198)*
(A:c) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
(AB:f) King Co. 902 N. Cedar St., Owatonna, Minn.
(A,c) Kohlenberger Engrg. Corp. 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
(A,B:d) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(A,B:b,c) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
(A,B:e) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
(B) Read Machinery Div., Standard Stoker Co., Inc., York, Pa.
(A,B:a,b,c,d) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B:d) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill. *(p. 201)*
(A,B:a,b,c,d) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas, St., Canton 4, O. *(p. 197)*
(A,B,a,d) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(A,B:c,d) Rigidbilt, Inc., 2850 W. Fulton St., Chicago 12, Ill.
(A,B:d) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(A,B:a,b,c,d,e,f,g) Super-Cold Corp. 1020 E. 59th St. Los Angeles 1, Cal.
(g) Surface Combustion Corp., 2375 Dorr St., Toledo 1, O.
(B:g) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B:a,c) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
(c,d,e) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
(A,B:a,c) XL Refrigorating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B:c) York Corp., York, Pa. *(p. 52)*

UNIT COOLERS BELOW 32°—WALL MOUNTED

(A—Ammonia; B—Other refrigerants)
(a—Reverse cycle; b—Electric; c—Brine spray; d—Water; e—Glycol; f—Warm air; g—Other)

(B:b) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B:a,c,e) Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
(B:a,b,d) Betz Corp., 445 State St., Hammond, Ind.
(A:c) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
(A,B:a,b,c,d) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
(A,B:a,b,c,d) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(A,B:f) King Co., 902 N. Cedar St., Owatonna, Minn.
(A,c) Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
(A,B:d) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 199)*
(A,B:b) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 200)*
(A,B) D. J. Murray Mfg. Co., 1024-3rd St., Wausau, Wis.
(A,B:e) Niagara Blower Co., 405 Lexington Ave., N.Y.C. 17 *(p. 97)*
(A,B:a,b,c,d) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B:a,b,d) Refrigeration Appliances, Inc., 917 W. Lake St. Chicago 7, Ill. *(p. 201)*
(A,B:a,b,d) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
(A,B:d) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 196)*
(A:d) Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.
(A,B:a,b,c,d,e,f,g) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(g) Surface Combustion Corp., 2375 Dorr St., Toledo 1, O.
(B:d,f,g) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B:b,c,d,e,f) Vilter Mfg. Co., 224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
(A,B:a,c) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

UNIVERSAL JOINTS (See FLEXIBLE COUPLINGS)

V-BELTS (See BELTS & BELTING)

VALVE DISCS, FLAPPERS, REEDS, WAFERS, etc

Acme Industrial Co 205 N Laflin St Chicago 7 Ill
Automatic Temperature Control Co Inc 5212 Pulaski Ave Philadelphia 44 Pa
Chicago Seal Co 332 S Hoyne Ave Chicago 20 Ill
Delavan Mfg Corp 3009 6th Ave Des Moines 13 Ia
Detroit Stamping Co 418 Midland Ave Detroit 3 Mich
Henry Disston & Sons Inc Tacony Phila 35 Pa
Hammel Dahl Co 243 Richmond St Providence 3 R I
Leslie Co Valley Brook & Grant Ave Lyndhurst N J
Sandvik Steel Inc 111 8th Ave N Y C 11
J H H Voss Co 785 E 144th St N Y C 54

VALVE PACKINGS (See GASKETS)

VALVE POSITIONERS

Automatic Temperature Control Co Inc 5212 Pulaski Ave, Phila 44 Pa
Bristol Co Waterbury 20 Ct
Brown Instrument Co Div Minneapolis Honeywell Regulator Co 4414 Wayne Ave Phila 44 Pa
Conoflow Corp 2100 Arch St Phila 28 Pa
Fischer & Porter Co Hatboro Pa
Hammel Dahl Co 243 Richmond St Providence 3 R I
Johnson Service Co 507 E Michigan Ave Milwaukee 22 Wis
Mason Neilan Regulator Co 1190 Adams St Boston 24 Mass
Minneapolis-Honeywell Regulator Co, 2933-4th Ave, S, Minneapolis 8, Minn *(p [illegible])*
Moore Products Co H & Lycoming Sts Phila 24 Pa
Taylor Instrument Cos 95 Ames St Rochester 1 N Y
Wheelco Instruments Co Harrison & Peoria Sts Chicago 7 Ill

VALVES (See particular type under type name or following)

VALVES, AMMONIA LINE

Ashton Valve Co 43 Kendrick St Wrentham Mass
Baker Refrigeration Corp, S Windham, Me *(p [illegible])*
Crane Co, 836 Michigan Ave, Chicago 5, Ill *(p [illegible])*
Creamery Package Mfg Co, 1243 W Washington Blvd, Chicago 7, Ill *(p [illegible])*
Dersch, Gesswein & Neuert, Inc, 4845 W Grand Ave, Chicago 39, Ill *(p 4[illegible])*
Frick Co, Waynesboro, Pa *(p [illegible])*
Hammel Dahl Co 243 Richmond St Providence 3 R I
Henry Valve Co, Melrose Park, Ill *(p [illegible])*
Kerotest Mfg Co, 2525 Liberty Ave, Pittsburgh 22, Pa *(p [illegible])*
Klingerit Inc 16 Hudson St N Y C 13
Kohlenberger Engrg Corp 1600 W Commonwealth Fullerton, Cal *(p [illegible])*
Lunkenheimer Co Cincinnati 14 O
H A Phillips & Co 3255 W Carroll Ave, Chicago 24, Ill *(p 142)*
Cyrus Shank Co, 623 W Jackson Blvd, Chicago 6 Ill *(p [illegible])*
Vilter Mfg Co, 2224 S 1st St, Milwaukee 7, Wis *(p 33)*
Henry Vogt Machine Co 10th & Ormsby St Louisville 10 Ky
Watson-Stillman Co, Roselle N J *(p 111)*
Worthington Pump & Machinery Corp, Harrison, N J *(p 144)*
XL Refrigerating Co 1834 W 59th St Chicago 36 Ill
York Corp, York, Pa *(p [illegible])*

VALVES, ANGLE, GATE, GLOBE etc, for AIR STEAM, WATER, etc

A D Anderson Co 1935 W 96th St Cleveland 2 O
Ashton Valve Co, 43 Kendrick St Wrentham, Mass
Automatic Temperature Control Co Inc 5212 Pulaski Ave Phila 44 Pa
Bailey Meter Co 1050 Ivanhoe Rd Cleveland 10 O

(Continued)

HENRY

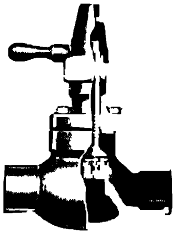
Balanced-Action Diaphragm
Packless Valves

BLUE BANTAM TYPE

Two-way line shut-off valves, flare or solder connection. Contain all the full proven features of STANDARD line except that diaphragm cannot be inspected or replaced while valves are under pressure. Positive seal between diaphragms and balancing channel with valve in closed position. Stock sizes $\frac{1}{4}$" thru [illegible]" flare and O D solder

STANDARD TYPE

Two-way, branch shut-off and angle types, flare or solder connections. Hand expansion, purge and charging types. Also available forged brass body and bonnet parts in line, non-directional. Four laminated diaphragms and stainless steel spring. [illegible] Stock sizes $\frac{1}{4}$" thru [illegible] flare [illegible] thru [illegible] O D solder $\frac{3}{4}$" thru [illegible]



WING CAP PACKED VALVES

Bronze with solder connection in globe and angle types sizes $\frac{7}{8}$" thru [illegible] O D. Semi-steel with F P T connection in globe and angle types sizes [illegible] thru 2". Semi-steel with bolted bonnets and square companion flanges with brass tailpieces (O D Solder) $1\frac{1}{8}$" thru $3\frac{1}{8}$". Flanged valves also available with steel tailpieces for welding to pipe [illegible] thru 3" I P S

Non-rotating replaceable swivel stem disc. Back seating to permit repacking under pressure. Wrench socket in top of wing cap to operate valve

WRITE FOR CATALOG

HENRY VALVE CO.

Melrose Park Illinois Suburb of Chicago

APPROVED FOR USE BY THE ARMY • NAVY • COAST GUARD AND THE MARITIME COMMISSION.

VALVES (Continued)

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Thomas Beckett & Co., Inc., P.O. Box 7354, Dallas 2, Tex.
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Conoflow Corp., 2100 Arch St., Phila. 28, Pa.
Consolidated Brass Co., 139 Summit, Detroit 9, Mich.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Edward Valves, Inc., Subsidiary of Rockwell Mfg Co., 1200 W. 145 St., E. Chicago, Ind.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 51)*
B. F. Goodrich Co., 500 S. Main St., Akron. O.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 203)*
Jenkins Bros., 80 White St., N.Y.C. 13
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Ludlow Valve Mfg. Co., Inc., Foot of Adams St., Troy, N.Y.
Lunkenheimer Co., Beckman St. & Waverly Ave., Cin'ti 14, O.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Manning, Maxwell & Moore, Inc., 11 Elias St., Bridgeport 2, Ct.
James P. Marsh Corp., Skokie, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 67)*
Mueller Co., 512 Cerro Gordo St., Decatur 70, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6 (Lead & Lead-lined)
W. H. Nicholson & Co., 209 Oregon St., Wilkes-Barre, Pa.
Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
C. A. Norgren, 222 Santa Fe Drive, Denver, Colo.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh 1, Pa.
Cyrus Shank Co., 623 W. Jackson Blvd., Chicago 6, Ill. *(p. 155)*
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 204)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y.
Trane Co., La Crosse, Wis. *(p. 118)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 111)*
White Flomatic Corp., P.O. Box 267, Hoosick Falls, N.Y.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*

VALVES, CARBON DIOXIDE

Bastian-Blessing Co., 4101 W. Peterson Ave., Chicago 40, Ill.
Conoflow Corp., 2100 Arch St., Phila. 28, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
York Corp., York, Pa. *(p. 52)*

VALVES, CHECK, AIR, WATER (See CHECK VALVES)

VALVES, CHECK, REFRIGERANT (See CHECK VALVES, REFRIGERANT)

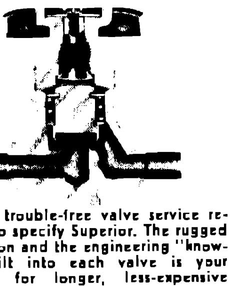
VALVES, COMPRESSOR SHUT-OFF

Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Henry Valve Co., Melrose Park, Ill. *(p. 203)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 107)*
Jenkins Bros., 80 White St., N.Y.C. 13
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 205)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Metrex Valve Co., 9900 Franklin Ave., Franklin Park, Ill.
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 204)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

VALVES, COMPRESSED GAS CYLINDER

Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg Co., Foot Walnut St., Williamsport, Pa.

SUPERIOR
Diaphragm Packless
LINE SHUT-OFF VALVES



For long, trouble-free valve service remember to specify Superior. The rugged construction and the engineering "know-how" built into each valve is your assurance for longer, less-expensive service. Write today for further information or our Catalog R-4.

SUPERIOR
valve & fittings co.
1509 W. Liberty Ave., Pittsburgh 26, Pa.

Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. (*p. 204*)

VALVES, EXPANSION (See EXPANSION VALVES)

VALVES, FLOAT, REFRIGERANT (See HIGHSIDE FLOATS; also LOWSIDE FLOATS)

VALVES, FLOAT, WATER (See FLOAT VALVES, WATER)

VALVES, FOOT

Jas. B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Conoflow Corp., 2100 Arch St., Phila. 28, Pa.
Crane Co., 836 S. Michigan Ave., Chicago 5, Ill. (*p. 109*)
Deming Co., 884 S. Broadway, Salem, O.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Jacuzzi Bros., Inc., 5327 Jacuzzi Ave., Richmond, Cal.
Jenkins Bros., 80 White St., N.Y.C. 13
Kennedy Valve Mfg. Co., Elmira, N.Y.
Ludlow Valve Mfg. Co., Inc., Foot of Adams St., Troy, N.Y.
A. Y. McDonald Mfg. Co., 12th & Pine Sts., Dubuque, Ia.
Swaby Mfg. Co., 3818 W. Armitage Ave., Chicago 47, Ill.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 12, Pa.

VALVES, MOTOR OPERATED

Barber-Colman Co., Rockford, Ill.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Bristol Co., Waterbury 20, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Conoflow Corp., 2100 Arch St., Phila. 28, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Cutler-Hammer, Inc., 315 N. 12 St., Milwaukee 1, Wis
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Grinnell Co., Inc., 260 W. Exchange St., Providence 1, R.I.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I
Jenkins Bros., 80 White St., N.Y.C. 13
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Ludlow Valve Mfg. Co., Inc., Foot of Adams St., Troy, N.Y.
Lunkenheimer Co., Beekman St., & Waverly Ave., Cin'ti. 14, O
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (*p. 67*)
W. H. Nicholson & Co., 209 Oregon St., Wilkes-Barre, Pa.
Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20 Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y
Walworth Co., 60 E. 42nd St., N.Y.C. 17

VALVES, NEEDLE (See EXPANSION VALVES, HAND OR NEEDLE)

VALVES, PRESSURE RELIEF (See PRESSURE RELIEF VALVES)

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VALVES, RECEIVER

Crane Co., 836 Michigan Ave., Chicago 5, Ill. (*p. 109*)
Frick Co., Waynesboro, Pa. (*p. 51*)
Henry Valve Co., Melrose Park, Ill. (*p. 208*)
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. (*p. 204*)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

VALVES, REFRIGERANT SHUT-OFF, NON-FERROUS

Automatic Switch Co., 41 E. 11th St., N.Y.C. 3
Chase Brass & Copper Co., 236 Grand St., Waterbury 20, Ct.
Fulton Sylphon Div. Robertshaw-Fulton Controls Co., Knoxville 4, Tenn.
General Controls Co., 801 Allen Ave., Glendale 1, Cal. (*p. 64*)
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. (*p. 208*)
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. (*p. 205*)
Mueller Brass Co., Port Huron, Mich.
Paxton-Mitchell Co., 2614 Martha St., Omaha 5, Neb.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. (*p. 204*)
Watson-Stillman Co., Roselle, N.J. (*p. 111*)

VALVES, REGULATING (See REGULATORS)

VALVES, SADDLE

Fairbanks Co., 393 Lafayette St., N.Y.C. 3
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. (*p. 107*)
Walworth Co., 60 E. 42nd St., N.Y.C. 17

VALVES, SOLENOID (See SOLENOID VALVES)

VAPOR SEALS

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Angier Corp., Framingham, Mass.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Lexington Supply Co., 4815 Lexington Ave., Cleveland 3, O.
Munn and Steele, Inc., 130 Lister Ave., Newark 5, N.J.
Sisalkraft Co., 205 W. Wacker Dr., Chicago 6, Ill.
Stancal Asphalt & Bitumuls Co., 200 Bush St., San Francisco 22, Cal.

VARIABLE SPEED TRANSMISSIONS (See SPEED CHANGERS)

VARNISHES, INSULATING

E. I. duPont deNemours & Co., Inc., Wilmington 98, Del.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Glidden Co., 11001 Madison Ave., Cleveland 2, O.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
A. C. Horn Co., Inc., 10th St. & 44th Ave., Long Island City 1, N.Y.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Midland Paint & Varnish Co., 3801 E. 91st St., Cleveland 5, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.

VEGETABLE REFRIGERATORS (See DISPLAY CASES)

VENDING MACHINES, REFRIGERATED

Artkraft Mfg. Corp., Kibby St. & D.T.&I.R.R., Lima, O.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Cole Products Corp., 39 La Salle St., Chicago 3, Ill.
General Vending Machine Corp., 549 W. Washington Blvd., Chicago 6, Ill.
Healthflow, Inc., Barnett Bank Bldg., Jacksonville 2, Fla.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Hupp Corp., Refrigeration Products Div., 1250 W. 76th St., Cleveland 2, O.
Interstate Engrg. Corp., 2250 E. Imperial Highwy., El Segundo, Cal.
O. D. Jennings & Co., 4309 W. Lake St., Chicago 24, Ill.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. (*p. 59*)
Revco, Inc., Deerfield, Mich.
Spacarb, Inc., 311 E. 23rd St., N.Y.C. 10
Vendorlator Mfg. Co., 4000 Railroad, Fresno, Cal.
Wilson Refrigeration Inc., S. Main St., Smyrna, Del.

VENEERS (See PLYWOOD)

VENTILATORS & VENTILATING SYSTEMS

Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
Air & Refrigeration Corp., 475 5th Ave., New York 17, N.Y.
Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
Allen Ventilator Div., Production Planning Co., 704 Woodward, Rochester, Mich.
American Blower Corp. Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Larson Ventilating Co., 1004 Keystone Bldg., Pittsburgh 22, Pa.
Arex Co., 333 N. Michigan Ave., Chicago 1. Ill.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
R. H. Bishop & Co., 103 N. 2nd St., Champaign, Ill.
Burt Mfg. Co., 932 S. High St., Akron 11, O.
Chelsea Fan & Blower Co., Inc., 639 South Ave., Plainfield, N.J.
Chicago Blower Corp., 4588 W. Congress St., Chicago 24, Ill.
De Bothezat Fan Div., American Machine & Metals, Inc., E. Moline, Ill.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
King Co., 902 N. Cedar St., Owatonna, Minn.
Knowles Mushroom Ventilator Co., 583 Upper Mountain Ave., Upper Montclair, N.J.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Maysteel Products, Inc., 740 N. Plankington Ave., Milwaukee 3, Wis.
Herman Nelson Div., American Air Filter Co., Inc., 1824-3rd Ave., Moline, Ill.
John J. Nesbitt, Inc., Phila. 36, Pa. (*p. 9*)
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.
Robbins & Myers, Inc. 1345 Lagonda Ave., Springfield, O.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Readville St., Boston 36, Mass. (*p. 23*)
O. A. Sutton Corp., 1812 W. 2nd St., Wichita 1, Kan.
Swartwout Co., 18511 Euclid Ave., Cleveland 12, O.
Trane Co., La Crosse, Wis. (*p. 118*)
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn (p. *37*)
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.

VENTILATORS, WINDOW

Acme Equip. Co., 205 E. B'way, Muskogee, Okla.
Burt Mfg. Co., 932 S. High St., Akron 11, O.
Chelsea Fan & Blower Co., Inc., 639 South Ave., Plainfield, N.J.
De Bothezat Fans Div., American Machine & Metals, Inc., E. Moline, Ill.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Paasche Airbrush Co., 1909 Diversey Pkwy., Chicago 14, Ill.
Piqua Machine & Mfg. Co., Young & Coolidge Sts., Piqua, O.
Pleasantaire Corp., 14th & K Sts., N.W., Washington 5, D.C.
Premium Plastics, Inc., 625 Salem Ave., Dayton 6, O.
Robbins & Myers, Inc., 1345 Lagonda Ave., Springfield, O.
O. A. Sutton Corp., 1812 W. 2nd St., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 118)*
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.

VESSELS, PRESSURE (See PRESSURE VESSELS)

VIBRATION ABSORBERS, LINE

Allied Metal Hose 37-46 9th St., Long Island City, N.Y.
American Brass Co., Waterbury 88, Ct. *(p. 189)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Atlantic Metal Hose Co., Inc., 123 W. 64th St., N.Y.C. 23
Barco Mfg. Co., 1801 Winnemac Ave., Chicago 40, Ill.
Chicago Metal Hose Corp., Maywood, Ill.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Finn & Co., 2850-5th Ave., N.Y.C. 30
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Grinnell Co., Inc., 260 W Exchange St., Providence 1, R.I.
Johnson Metal Hose Co., 226 Mill St., Waterbury 88, Ct.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N Y.
Techniflex Corp., 53 Jersey St., Port Jervis, N.Y.
Titeflex, Inc., 300 Frelinghuysen Ave., Newark 5, N.J.

VIBRATION ABSORBING BASES & MOUNTINGS

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Armstrong Cork Co., Lancaster, Pa. *(p. 132)*
Barry Corp., 700 Pleasant St. Watertown 72, Mass.
Beltron Associates, Inc., 225 Lafayette St., N.Y.C. 12
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Bushings, Inc., 4358 Coolidge Highwy., Royal Oak, Mich
Cork Import Corp., 1 Engle St., Englewood, N.J.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17
Delgro Mfg. Co., Inc., Boonton, N.J.
Felters Co., 210 South St., Boston 1, Mass.
Finn & Co., 2850-5th Ave., N.Y.C. 30
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 171)*
L. H. Gilmer Co., Div. of U. S. Rubber Co., 7230 Keystone St., Tacony, Phila., 35, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 135)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11
Korfund Co., Inc., 48-51-32nd Place, Long Island City 1, N.Y.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
MB Mfg. Co., Inc., 1060 State St., New Haven 11, Ct.
Mundet Cork Corp., 7101 Tonnelle Ave., North Bergen, N.J. *(p. 138)*
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Rector Mineral Trading Corp., 16 E. 43rd St., N.Y.C. 17 *(p. 136)*
Rubatex Div., Great American Industries, Bedford, Va. *(p. 130)*
Sponge Rubber Products Co., 1600 Derby Place, Shelton, Ct.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.
Vibration Eliminator Co., 10-28-47th Ave., Long Island City 1, N.Y.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

VOLTMETERS (See METERS, ELECTRIC)

WALK-IN COOLERS

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

Allied Store Equip. Co., 116 N. 7th St., Minneapolis, Minn.
Amana Refrigeration Inc., Amana Ia.
(A,B) American Refrigeration Corp., 1025 E. Excelsior Ave., Hopkins, Minn.
(A,B) Annapolis Yacht Yard, Box 791, Annapolis, Md. *(p. 207)*
(C) Bally Case & Cooler Co., Bally, Pa.
(A,B,C) Bowser Technical Refrigeration Div. Bowser, Inc. Terryville, Ct.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
(C) Commercial Cooler Mfg. Co., 5736-12th, St., Detroit, 8, Mich.
(A,B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B,C) Delaware Refrigeration Co., 834 N. 6th St., Phila 23, Pa.
Delgro Mfg. Co., Inc., Boonton, N.J.
(A,C) Esco Cabinet Co., West Chester, Pa.

(Continued)

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WALK-IN COOLERS (Continued)

(A,B,C) Evans Mfg. Corp., 460 S. 10th Ave., Mt. Vernon, N.Y.
(A,B,C) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B) Ed Friedrich, Inc., 1117 E. Commerce St., San Antonio 6, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
(A,B,C) Gem Refrigerator Co., 165 W. Wyoming Ave., Phila. 40, Pa.
Harder Refrigerator Div., Tyler Fixture Corp., Cobleskill, N.Y.
(A,B,C) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. (*p. 165*)
(A,B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 83*)
(A,B,C) Jordon Refrigerator Co., 58th St. & Gray's Ave., Phila. 43, Pa.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
(A,B,C) Lingle Refrigerator Co., Inc., 1116 E. 15th St., Kansas City 6, Mo.
(B) McCray Refrigerator Co., Kendallville, Ind
(A,B,C) Masterfreeze Corp., Sister Bay, Wis.
Minneapolis Show Case & Fixture Co., New Richmond, Wis.
(A,B) Morton Show Cases, Inc., Washington Courthouse, O.
(A,B,C) John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
(A,B,C) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(A,B,C) Nor-Lake, Inc., 2nd & Elm Sts., Hudson, Wis.
(C) R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
(A,B,C) J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
(B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(A,B,C) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(B,C) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. (*p. 82*)
(A,C) Uniflow Mfg. Co., Erie, Pa.
(A,B,C) United Refrigerator Co., Hudson, Wis.
(B) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
(A,B,C) Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
(B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B,C) Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.
(B,C) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(A,B,C) Wilson Refrigeration Inc., S. Main St., Smyrna, Del.

WALK-IN FREEZERS (See also FARM & HOME FREEZERS)

Amana Refrigeration Inc., Amana, Ia.
American Refrigeration Corp., 1025 E. Excelsior Ave., Hopkins, Minn.
Bowser Technical Refrigeration Div. of Bowser Inc., Terryville, Ct.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. (*p. 25*)
Commercial Cooler Mfg. Co., 5736-12th, Detroit 8, Mich.
Delaware Refrigeration Co., 834 N. 6th St., Phila. 23, Pa.
Delgro Mfg. Co., Inc., Boonton, N.J.
Esco Cabinet Co., West Chester, Pa.
Evans Mfg. Corp., 460 S. 10th St., Mt. Vernon, N.Y.
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 3, Tex.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
Gem Refrigerator Co., 165 W. Wyoming Ave., Phila. 40, Pa.
Harder Refrigerator Div., Tyler Fixture Corp., Cobleskill, N.Y.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Hussmann Refrigerator Co., 2401 N. Leffingwell, St. Louis 6, Mo. (*p. 83*)
Jordon Refrigerator Co., 58th St. & Gray's Ave., Phila. 43, Pa.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Masterfreeze Corp., Sister Bay, Wis.
Minneapolis Show Case & Fixture Co., New Richmond, Wis.
John Mowat Refrigerators, 1866 Folsom St., San Francisco 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
C. Schmidt Co., 1712 John St., Cincinnati, O.
Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
United Refrigerator Co., Hudson, Wis.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Ward Refrigerator & Mfg. Co., 6501 S. Alameda St., Los Angeles 1, Cal.
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
Wilson Refrigeration Inc., S. Main St., Smyrna, Del.
York Corp., York, Pa. (*p. 52*)

WASHERS

Allmetal Screw Products Co., Inc., 33 Greene St., N.Y.C. 13
American Nut & Bolt Fastener Co., 2029 Doerr St., Pittsburgh 12, Pa.
Armstrong Cork Co., Lancaster, Pa. (*p. 132*)
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J
Bossert Co., P.O. Drawer 358, Utica 1, N.Y.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
A. W. Chesterton Co., 4 Ashland St., Everett 49, Mass.
Clark Bros. Bolt Co., Milldale, Ct.
Continental Diamond Fibre Co., 3 Chapel St., Newark, Del.
Detroit Stamping Co., 418 Midland Ave., Detroit 3, Mich.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Goshen Rubber Co., Inc., Box 517, Goshen, Ind.
Graton & Knight Co., 356 Franklin St., Worcester 4, Mass.
H. M. Harper Co., 8200 Lehigh Ave., Morton Grove, Ill.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Lakeside Malleable Castings Co., Racine, Wis.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
New York Belting & Packing Co., 1 Market St., Passaic, N.J.
Philadelphia Steel & Wire Corp., 5250 Belfield Ave., Phila. 44, Pa.
Phoenix Specialty Mfg. Co., Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Premium Plastics, Inc., 627 Salem Ave., Dayton 6, O.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.

(*Continued*)

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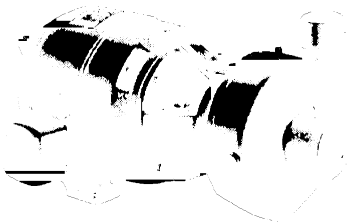
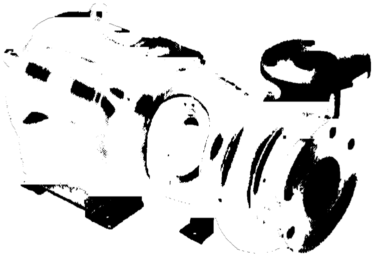
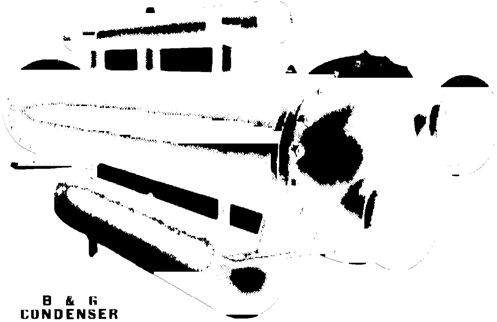


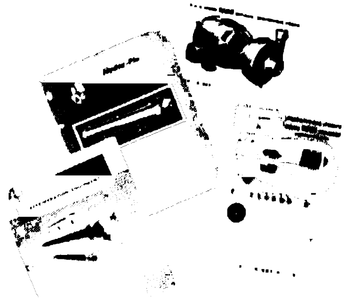
B & G SERIES 1522 CENTRIFUGAL PUMP



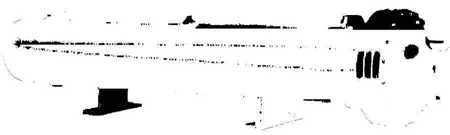
B & G SERIES 1531 CENTRIFUGAL PUMP



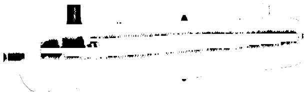
B & G CONDENSER



Send for catalog giving descriptions and engineering data. You'll agree that here is refrigeration equipment you can depend upon



B & G EVAPORATOR



B & G HEAT EXCHANGER

BELL & GOSSETT COMPANY

Morton Grove, Illinois

WASHERS (Continued)

Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Standard-Keil Hardware Mfg. Co., Inc., 639 Broadway, N.Y.C. 12
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Thompson-Bremer & Co., 1610 W. Hubbard St., Chicago 22, Ill.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16, Mich.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

WASHERS, AIR (See AIR WASHERS)

WASHERS, RUBBER (See RUBBER PRODUCTS)

WATER COILS (See AIR CONDITIONING COILS)

WATER COOLERS (See particular type; also DRINKING WATER COOLERS)

WATER COOLERS, ATMOSPHERIC (See COOLING TOWERS)

WATER COOLERS, BAUDELOT TYPE

Air & Refrigeration Corp. 475 5th Ave., New York 17, N.Y.
Baker Refrigeration Corp., S. Windham, Me. *(p. 63)*
Burge Ice Machine Co., 218 N. Jefferson St., Chicago 6, Ill.
Cherry Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Chester-Jensen Co., Fifth & Tilghman Sts., Chester, Pa.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Nooter Corp., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

WATER COOLERS, DRY EXPANSION

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. *(p. 209)*
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
Drayer Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 90)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 17)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Heat-X-Changer Co., Inc., Brewster, N.Y. *(p. 120)*
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 4)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
Larkin Coils, Inc., 519 Memorial Dr. S.E. Atlanta, Ga. *(p. 41)*
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Read Machinery Div., Standard Stoker Co., Inc., York, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Vilter Mfg. Co., 2223 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
XL Refrigerating Co., 1834 W. 59th St., Chicago 37, Ill.

WATER COOLERS, SHELL & COIL

Acme Industries, Inc., Mechanic & Ganson Sts. Jackson, Mich. *(p. 211)*
Richard M. Armstrong Co., Box., 188, W. Chester, Pa.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. *(p. 209)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 31)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 90)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Interstate Engrg. Corp., 2250 E. Imperial Highwy., El Segundo, Cal.
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich. *(p. 21)*
Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. *(p. 39)*
L. O. Koven & Brother, Inc., 154 Ogden Ave., Jersey City 7, N.J.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Potter & Rayfield, Inc., P.O. Box 1042, Atlanta 1, Ga.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 197)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Edw. Renneburg & Sons Co., 2639 Boston St., Baltimore 24, Md.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Star Metal Mfg. Co., Inc., Trenton Ave., & Ann St. Phila. 34, Pa.
Taco Heaters Inc., 137 South St., Providence 3, R.I.
Temprite Products Corp., E. Maple Rd., Birmingham, Mich. *(p. 89)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 33)*
Whitlock Mfg. Co., 81 South St., Hartford 10, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 144)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 52)*

WATER COOLERS, SHELL & TUBE

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 211)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Baker Refrigeration Corp., Inc., S. Windham, Me. *(p. 63)*
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. *(p. 209)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 25)*
Creamery Package Mfg., Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 49)*
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Doyle & Roth Mfg. Co., Foot 23rd St., Brooklyn 32, N.Y. *(p. 51)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 90)*
Frick Co., Waynesboro, Pa. *(p. 51)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.

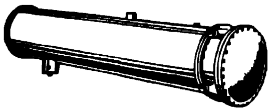
(Continued)

ACME INDUSTRIES, INC.

Jackson, Michigan

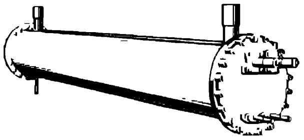
Representatives in Principal Cities

Continuously Serving The Refrigeration Industry Since 1919



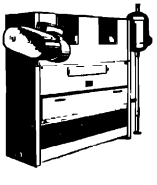
FREON CONDENSERS AMMONIA CONDENSERS

Shell and Tube type for use with Ammonia, Freon or other Refrigerants. Standard or special designs to meet varying water temperatures available and condensing temperatures desired.



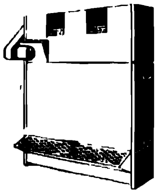
DRY-EX COOLERS

Refrigerant in Tubes, solution baffled through shell. For cooling water, brine, glycols or alcohols by direct expansion of refrigerant



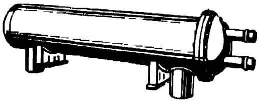
BLO-COLD INDUSTRIAL UNIT COOLERS

Blo-Cold Models are available for either medium temperature or low-temperature applications. *All fabricated steel parts are hot-dip galvanized after fabrication.*



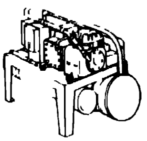
EVAPORATIVE CONDENSERS AND COOLING TOWERS

All prime surface for Freon or Ammonia Refrigerants—Heavy gage sheet metal casings, especially processed for maximum resistance to rust and corrosion Capacities to 100 tons. *All fabricated steel parts are now hot-dip galvanized after fabrication*



HEAT INTERCHANGERS

Shell and Coil units for small capacities, shell and tube units for large installations 16 standard models from one ton to 180 tons capacity.



FLOW-COLD LIQUID CHILLERS

A *complete*, compact unit for Industrial Process Temperature Control, Air Conditioning and Heat Pump applications. Factory Assembled and tested, from 2 to 15 H.P.

Acme Industries, Inc. also manufacture Flooded Water and Brine Coolers, Oil Separators, Pipe Coils, Heat Exchangers, Convertors, Receivers and Fin Coils

WRITE FOR CATALOG ON ANY PRODUCT

WATER COOLERS (Continued)

Kohlenberger Engrg. Corp., 1600 W. Commonwealth, Fullerton, Cal. (*p. 39*)
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. (*p. 195*)
Lummus Co., 420 Lexington Ave., N.Y.C.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Mo. (*p. 75*)
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. (*p. 58*)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. (*p. 197*)
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
B. F. Sturtevant Div., Westinghouse Elec. Corp., 101 Reading St., Boston 36, Mass. (*p. 23*)
Taco Heaters Inc., 137 South St., Providence 3, R.I.
Trane Co., La Crosse, Wis. (*p. 118*)
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 51, N.Y.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. (*p. 33*)
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. (*p. 52*)

WATER GAUGE COLUMNS (See GAUGE GLASSES)

WATER HEATERS

Airtemp Div., Chrysler Corp., 1600 Webster, Dayton 1, O. (*p. 19*)
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Richard M. Armstrong Co., Box 188, W. Chester, Pa.
Bell & Gossett Co., 8200 Austin Ave., Morton Grove, Ill. (*p. 209*)
Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.
Davis Engrg. Corp., 1064 E. Grand St., Elizabeth 4, N.J.
Electric Heater Co., Woodend Rd., Stratford, Ct.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago, 5. Ill.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. (*p. 17*)
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Heat-X-Changer Co., Inc., Brewster, N.Y. (*p. 120*)
Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich (*p. 21*)
Patterson-Kelley Co., E. Stroudsburg, Pa. (*p. 58*)
Rheem Mfg. Co., 570 Lexington Ave., N.Y.C. 22
Servel, Inc., Evansville 20, Ind.
A. O. Smith Corp., 3533 N. 27th St., Milwaukee 1, Wis.
David Stout & Sons, 7016 Manchester Blvd., St. Louis 17, Mo.
Taco Heaters Inc., 137 South St., Providence 3, R.I.
Whitlock Mfg., Co., 81 South St., Hartford 10, Ct.
Wilson Cabinet Co., Inc., Smyrna, Del.

WATER STRAINERS (See STRAINERS)

WATER & BRINE TREATING & TESTING EQUIPMENT & MATERIALS

American Sand-Banum Co., Inc., 9 Rockefeller Plaza, N.Y.C. 20
American Water Softener Co., S.E. Corner 4th & Lehigh Ave., Phila. 33, Pa.
Aquatrol Laboratories, Inc., Clifton, N.J.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Henry Bower Chemical Mfg. Co., Gray's Ferry Rd. & 29th St., Phila. 46, Pa.
Calgon, Inc., Hagan Bldg., Pittsburgh 30, Pa.
Carbide & Carbon Chemicals Corp., Div. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Dearborn Chemical Co., Merchandise Mart Plaza, Chicago 54, Ill.
Dow Chemical Co., Midland, Mich.
Elgin Softener Corp., Elgin, Ill.
Ferro-Nil Corp., 381-4th Ave., N.Y.C. 16
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. (*p. 90*)
Graver Water Conditioning Co., 216 W. 14th St., N.Y.C 11
D. W. Haering & Co., P.O. Box 6037, Harlandale Station, San Antonio, Tex.
Hungerford & Terry, Inc., 226 Atlantic Ave., Clayton, N.J.
Infilco, Inc., 2750 S. 12th Ave., Tucson, Ariz.
Mathieson Chemical Corp., Mathieson Bldg., Baltimore 3, Md.
Mutual Chemical Co. of America, 270 Madison Ave., N.Y.C. 16
National Aluminate Corp., 6216 W. 66th Place, Chicago 38, Ill.
New Jersey Zinc Co., 160 Front St., N.Y.C. 7
North American Mogul Products Co., Standard Bldg., Cleveland 13, O.
Nox-Rust Chemical Corp., 2429 S. Halstead St., Chicago 8, Ill.
Oakite Products, Inc., 22 Thames St., N.Y.C. 6
O'Brien Industries, 84 Bishop St., Jersey City 4, N.J.
Oshkosh Filter & Softener Co., 51 Ceape St., Oshkosh, Wis.
Permutit Co., 330 W. 42nd St., N.Y.C. 18
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Milton Roy Co., 1401 E. Mermaid Ave., Phila. 18, Pa.
Sta-Rite Products, Inc., Delavan, Wis.
F. J. Stokes Machine Co., Tabor Rd., Phila. 20, Pa.
Tempo Chemical Co., Inc., 47-02 5th St., Long Island City 1, N.Y.
Tenney Engrg., Inc., 26 Ave. B., Newark 5, N.J.
Water Service Labs., Inc., 423 W. 126th St., N.Y.C. 27
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Wright Chemical Corp., 627 W. Lake St., Chicago 6, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. (*p. 144*)
York Corp., York, Pa. (*p 52*)

WATTMETERS (See METERS, ELECTRIC)

WEATHER STRIP (See also DOOR GASKETS)

Brasco Mfg. Co., Harvey, Ill.
Bridgeport Fabrics Co., Bridgeport 1, Ct.
Central Metal Strip Co., 4343 North Western Ave., Chicago 18, Ill.
Chamberlin Co. of America, 1254 La Brosse St., Detroit 26, Mich.
W. J. Dennis & Co., 1732 N. Kolmar Ave., Chicago 39, Ill.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hewitt Rubber Div., Hewitt-Robins, Inc., 240 Kensington Ave., Buffalo 5, N.Y.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. (*p. 85*)
Koch Butchers' Supply Co., 600 E. 14th Ave., N., Kansas City 16, Mo.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

WELDING ELECTRODES, RODS & WIRE

Air Reduction Sales Co., 60, E. 42nd St., N.Y.C. 17.
All-State Welding Alloys Co., Inc., 273 Ferris Ave., White Plains, N.Y.
American Brass Co., Waterbury 88, Ct. (*p. 189*)
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Arcos Corp., 1500 S. 50th St., Philadelphia 45, Pa.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Driver-Harris Co., Harrison, N.J.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Handy & Harman, 82 Fulton St., N.Y.C. 7
Hobart Bros. Co., 146 Hobart Square, Troy, O.
International Nickel Co., 67 Wall St., N.Y.C. 5 (Monel, Nickel, etc.)
Lincoln Elec. Co., 2818 Colfax Rd., Cleveland 1, O.
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind. (Electrodes only)
Marquette Mfg. Co., Inc., 307 E. Hennepin, Minneapolis 13, Minn.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 191)*
Rex Welder & Engrg. Co., 1316 E. 10th St., Kansas City 6, Mo.
A. O. Smith Corp., 3533 N. 21st St., Milwaukee 1, Wis.
Trindl Products, Ltd., 17 E. 23rd St., Chicago 16, Ill.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

WELDING ELECTRODE COOLERS (See ELECTRODE COOLING SYSTEMS)

WHEELS, BLOWER (See BLOWERS)

WINDOWS, COLD STORAGE (See COLD STORAGE DOORS & WINDOWS)

WIRE, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
Permanente Products Co., 1924 B'way, Oakland 12, Cal.
Reynolds Metals Co., 2000 S. 9th St., Louisville 1, Ky.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

WIRE, BRASS, BRONZE, COPPER, etc.

American Brass Co., Waterbury 88, Ct. *(p. 189)*
Whitney Blake Co., 1565 Dixwell Ave., New Haven 14, Conn.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cornish Wire Co., Inc., Rutland, Vt.
C. G. Hussey & Co., 2860 2nd Ave., Pittsburg 19, Pa.
Koiled Kords, Inc., 1565 Dixwell Ave., New Haven 14, Ct.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

WIRE, CABLE & ROPE

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Macwhyte Wire Rope Co., 2923-14th Ave., Kenosha, Wis
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts , Chicago, Il
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

WIRE, ELECTRIC (See also CABLE, ELECTRIC; CORDS, ELECTRIC)

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Anaconda Wire & Cable Co., 25 Broadway, N.Y.C.
Belden Mfg. Co., 4647 W. Van Buren St., Chicago, Ill.
Whitney Blake Co., 1565 Dixwell Ave., New Haven 14, Conn.
William Prand and Co., Inc , Willimantic, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cornish Wire Co., Inc., Rutland, Vt.
Federal Telephone & Radio Corp., 900 Passaic Ave., E. Newark, N. J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Instrument & Gauge Div., Electric Auto-lite Co., Champlain St., Toledo 1, O
Koiled Kords, Inc., 1565 Dixwell Ave., New Haven 14, Ct
National Elec. Products Corp., Chamber of Commerce Bldg., Pittsburgh 19, Pa.
Packard Elec. Div., Gen'l. Motors Corp., Warren, O.
Pent Elec. Co., Inc., 634 Michigan Trust Bldg., Grand Rapids 2, Mich.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Triangle Conduit & Cable Co., Inc., New Brunswick, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

WIRE, ELECTRIC RESISTANCE

Driver-Harris Co., Harrison, N.J.
General Electric Co., River Rd., Schenectady 5, N.Y

WIRE, GALVANIZED

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

WIRE, INCONEL, MONEL & NICKEL

Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
Claud S. Gordon Co., 3000 S. Wallace St., Chicago 16, Ill.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
United Wire & Supply Corp., 1497 Elmwood Ave., Providence, R.I.

WIRE, STEEL

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Armco Steel Corp., Middletown, O.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Bethlehem Steel Co., Bethlehem, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Sandvik Steel, Inc., 111-8th Ave., N.Y.C. 11.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp., Subsidiary, Brown-Marx Bldg., Birmingham, Ala
Youngstown Sheet & Tube Co., Youngstown, O.

WIRE, STAINLESS STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Arcos Corp., 1500 S. 50th St., Philadelphia 45, Pa.
Armco Steel Corp., Middletown, O.

(Continued)

WIRE, STAINLESS STEEL (Continued)

Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
Firth-Sterling Steel & Carbide Corp., Demmler Rd., McKeesport, Pa.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Sandvik Steel, Inc., 111-8th Ave., N.Y.C. 11
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary Brown Marx Bldg., Birmingham, Ala.

WIRE, TINNED

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Bethlehem Steel Co., Bethlehem, Pa.
William Brand and Co., Inc., Willimantic, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Cornish Wire Co., Inc., Rutland, Vt.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.

WIRE, WELDING (See WELDING ELECTRODES)

WIRE CLOTH & MESH & SCREENS

Bethlehem Steel Co., Bethlehem, Pa.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Cambridge Wire Cloth Co., Cambridge, Md.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Central Wire & Iron Wks., 621 E. Locust St., Des Moines 9, Ia.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Industrial Wire Cloth Products Corp., Wayne, Mich. (p. 88)
Kentucky Metal Products Co., Preston St. & Audubon Pk., Louisville 13, Ky.
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. (p. 86)
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Multi-Metal Wire Cloth Co., 1350 Garrison Ave., Bronx 59, N.Y.
Newark Wire Cloth, 351 Verona Ave., Newark 4, N.J.
Robins Conveyors Div., Hewitt-Robins Inc., 270 Passaic Ave., Passaic, N.J.
Rolock, Inc., 1350 Kings Highway, Fairfield, Ct.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
Wheeling Corrugating Co., Wheeling, W. Va.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

WIRE FORMING, WIRE SPECIALTIES & WIREWORK (See also SHELVES)

American Spring & Wire Specialty Co., 816 N. Spaulding Ave., Chicago 51, Ill.
Armstrong Co., 34-36 Bell Blvd., Bayside, N.Y.
Atlantic Steel Co., P.O. Box 1714, Atlanta 1, Ga.
Bauer Bros. Co., 1774 Sheridan Ave., Springfield, O.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Central Wire & Iron Wks., 621 E. Locust St., Des Moines 9, Ia.
Collis Co., Box 231, Clinton, Ia. (p. 174)
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Hunter Pressed Steel Co., 801 Maple St., Lansdale, Pa.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
Newark Wire Cloth, 351 Verona Ave., Newark 4, N.J.
Peerless Wire Goods Co., Lafayette, Ind.
Rolock, Inc., 1350 Kings Highway, Fairfield, Ct.
Scovill Mfg. Co., 61 Mill St., Waterbury 20, Ct.
Standard Keil Hardware Mfg. Co., Inc., 639 B'way, N.Y.C. 12
E. H. Titchener & Co., Binghamton, N.Y.
Townsend Co., 205 River Rd., New Brighton, Pa.
Union Steel Products Co., 448 Pine St., Albion, Mich. (p. 177
United Steel & Wire Co., Battle Creek, Mich.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

WIRING DEVICES & HARNESSES

Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.
Whitney Blake Co., 1565 Dixwell Ave., New Haven 14, Ct.
Cornish Wire Co., Inc., Rutland, Vt.
General Electric Co., 1 River Rd., Schenectady 5, N.Y.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Graybar Elec. Co., Inc., Graybar Bldg., N.Y.C. 17
Koiled Kords, Inc., 1565 Dixwell Ave., New Haven 14, Ct.
Packard Elec. Div., Gen'l Motors Corp., Warren, O.
Pent Elec. Co., Inc., 634 Michigan Trust Bldg., Grand Rapids 2, Mich.
Riverside Mfg. & Electrical Supply Co., 10221 Michigan Ave., Dearborn, Mich. (p. [illegible])
Soreng Mfg. Co., 9555 Eden Ave., Schiller Park, Ill.
United Mfg. & Service Co., 405 W. 6th, Milwaukee, Wis.

X-RAY COOLING EQUIPMENT (See PHOTOGRAPHIC COOLING EQUIPMENT)

ZINC

Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Illinois Zinc Co., 2959 W. 47th St., Chicago 32, Ill.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
National Lead Co., 111 Broadway, N.Y.C. 6
New Jersey Zinc Co., 160 Front St., N.Y.C. 7
Northwest Lead Co., 2700-16th Ave., S.W., Seattle 4, Wash.

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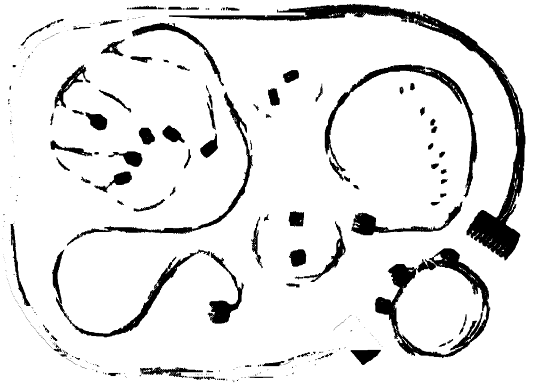

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